

GAS CONDITIONING AND PROCESSING

Volume 2: The Equipment Modules

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Volume 2 of "Gas Conditioning and Processing" is a continuation of Volume 1, "The Basic Principles." These two volumes are a single book published in two parts.

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ACKNOWLEDGMENT

This Seventh Edition represents a substantial rewrite and reorganization of the previous edition. This was done to reflect what I regard as necessary basic knowledge to function effectively in the subject area now and in the 1990's.

About thirteen revisions have occurred since this book had its birth as an adult training manual in 1960. Scores of folks have contributed to these changes by their formal or informal comments. Where possible, these have been cited. The source of some changes has been lost. I both apologize for this and thank these unknown persons for their contribution.

Special thanks are due Bob Hubbard, Larry Lilly, John Morgan, and Bob Maddox for their contributions in the preparation and review of the technical material. Sharon Antram is responsible for much of the new artwork. Joy Garrison (retired) is remembered for the many hours dedicated since 1968 to this and previous editions. Danny Stowe deserves special recognition for countless hours of wordprocessing, layout and editing work. His efforts pulled this manuscript together as a book. Thank you all for your help.

Dr. John M. Campbell

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10

FLOW OF FLUIDS

This chapter is an overview of a more detailed discussion in the book *Petroleum Fluid Flow Systems*, also published by Campbell Petroleum Series.^(10.1)

The flow of any fluid through a line containing no heat input or output may be considered isothermal and adiabatic. One may therefore combine the First and Second Laws of Thermodynamics, using these assumptions to write an equation.

$$\int V dP + \frac{g}{g_c} \Delta X + \frac{(\Delta v)^2}{2 g_c} = -W_f - W \quad (10.1)$$

Where: V = volume of the fluid
 P = pressure of the fluid
 ΔX = change in elevation of the fluid
 Δv = change in velocity of the fluid
 W_f = work lost due to friction
 W = work done by the system
 g = gravitational force
 g_c = mass-force conversion constant

Equation 10.1 is as far as one can go using thermodynamics alone, for it provides no way of solving for the term W_f , lost work.

The total irreversible effects (W_f) are usually attributed to friction. The work done in overcoming friction through a distance dL is proportional to the surface in contact with the fluid, approximately proportional to the square of velocity and proportional to the fluid density. By including a proportionality constant, f , the equation may be written.

$$\text{frictional resistance} = (f)(dL)(\pi d) \left(\frac{v^2}{2 g_c} \right) (\rho) \quad (10.2)$$

where the definition of variables for Equations 10.2-10.4 may be found on page 4.

The weight of fluid in the pipe is the length dL multiplied by the cross-sectional area and the fluid density. Any frictional work would be represented by the frictional resistance moved through distance dL . Combining these concepts into Equation 10.2 then yields dW_f :

$$dW_f = \frac{(f)(dL)(\pi d) \left(\frac{v^2}{2 g_c} \right) (\rho)(dL)}{(\pi/4)(d^2)(\rho)(dL)} \quad (10.3)$$

Simplifying:

$$dW_f = \frac{2 f v^2 dL}{g_c d} \quad (10.4)$$

which integrated between the limits of 0 to W_f and 0 to L gives

$$W_f = \frac{2 f L v^2}{g_c d} \quad (10.5)$$

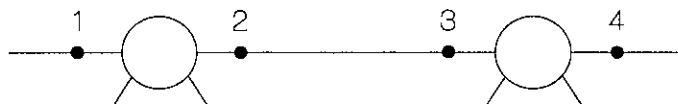
Equation 10.5 is called the *Fanning friction factor* equation. Other forms of this equation have been published which differ only in the value of the coefficient ahead of "f." In this book the Fanning equation will be used in all correlations.

Combination of Equations 10.1 and 10.5 then gives

$$\int V dP + \frac{g}{g_c} \Delta X + \frac{(\Delta v)^2}{2 g_c} = - \frac{2 f L v^2}{g_c d} - W \quad (10.6)$$

which is the basic equation for flow often called the Bernoulli equation.

When solving a problem for a line containing equipment like pumps, the line is divided into sections as shown below.



The pump sections (1-2 and 3-4) are solved by the equation

$$\int V dP = -W_{\text{theor.}} \quad (10.7)$$

The potential energy, kinetic energy and friction drop terms are incorporated into a single efficiency term (E) that replaces these terms.

For a pump,

$$W_{\text{actual}} = \frac{W_{\text{theor.}}}{E}$$

For a hydraulic expander,

$$W_{\text{actual}} = (E)(W_{\text{theor.}})$$

For the line section 2-3 no work is done. So, W is eliminated from an equation like 10.6 to solve for the pressure drop – flow rate behavior, giving Equation 10.8:

$$\int V dP + \frac{g \Delta X}{g_c} + \frac{\Delta v^2}{2 g_c} = - \frac{2 f L v^2}{g_c d} \quad (10.8)$$

EVALUATION OF FRICTION FACTOR

The friction factor "f" must be evaluated empirically. One common correlation is the Moody plot shown in Figure 10.1.^(10.2)

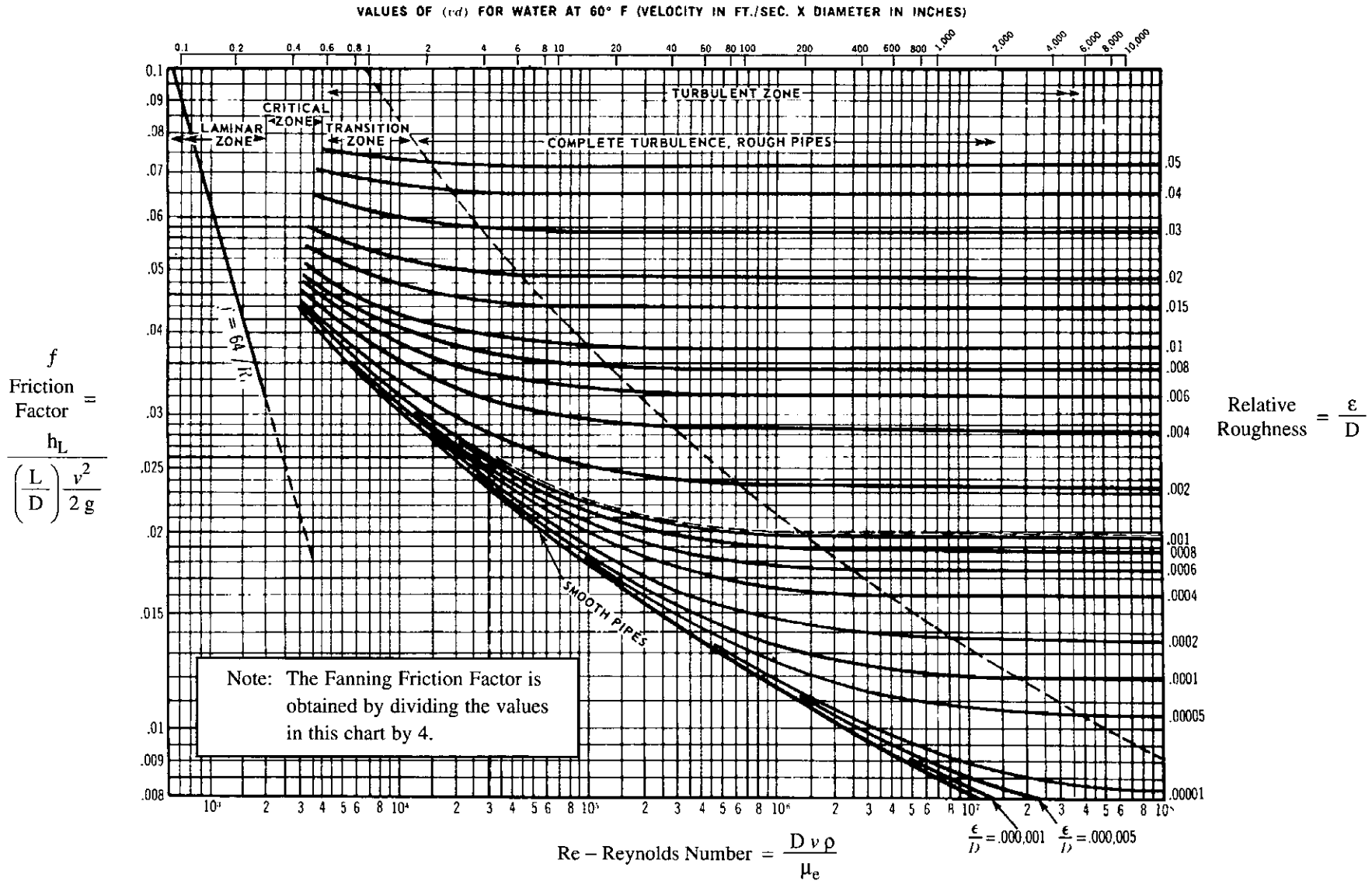


Figure 10.1 Friction Factors Using Moody Correlation^(10.4)
(which combines the Colebrook transition between the smooth pipe equation and the fully turbulent zone)

Factor "f" depends on pipe roughness and diameter, and the fluid involved. The dimensionless Reynolds number on the abscissa of Figure 10.1 may be written in various forms.

$$Re = \frac{d v \rho}{\mu} = \frac{q \rho}{0.785 d \mu} = \frac{m}{0.785 d \mu} \quad (10.9)$$

Any units that make Re dimensionless may be used. For fluid flow applications the following units are most common.

Where: d = internal line diameter
v = velocity
ρ = density
μ = viscosity
P = pressure
g_c = mass/force conversion constant

g = gravitational force
L = length of line
ΔX = change of elevation
m = mass flow rate
q = volumetric flow rate
f = dimensionless factor
0.785 = π/4
1 cp = 0.001 kg/m·s = 6.72 × 10⁻⁴ lb_m/ft·sec

Metric	English
m	ft
m/s	ft/sec
kg/m ³	lbm/ft ³
kg/m·s	lbm/ft·sec
Pa	lbf/ft ²
1.0 $\left(\frac{\text{kg} \cdot \text{m}}{\text{N} \cdot \text{s}^2} \right)$	32.17 $\left(\frac{\text{lbm} \cdot \text{ft}}{\text{lbf} \cdot \text{sec}^2} \right)$
9.81 m/s ²	32.17 ft/sec ²
m	ft
m	ft
kg/s	lbm/sec
m ³ /s	ft ³ /sec

This type of correlation applies only to *Newtonian fluids*. Gas and almost all liquids of concern in this book are Newtonian in nature.

Non-Newtonian Liquids

One often assumes (erroneously) that the liquids being used are Newtonian fluids. In fact, many fluids do not truly fall within this category. There are several general classifications of liquids based on their rheological behavior: *Newtonian*, *Bingham Plastic*, *Pseudoplastic* and *Dilatant*.

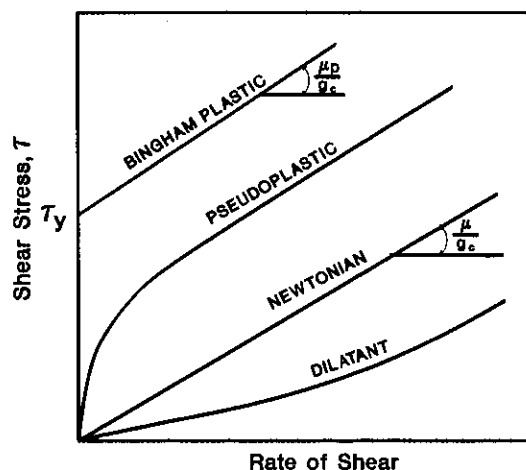


Figure 10.2 Shear Curves of Newtonian and Non-Newtonian Fluids

The general curves for these are shown in Figure 10.2.

Of these, the Bingham plastic and Newtonian liquids are the most common models employed. For a Newtonian liquid, the relationship is expressed by the equation

$$\tau = \frac{\mu}{g_c} \left(-\frac{dv_r}{dr} \right) \quad (10.10)$$

Where: μ = absolute viscosity
-(dv_r/dr) = shear rate for laminar flow in a circular pipe

As shown in Figure 10.2, this equation produces a straight line on cartesian coordinates starting at the origin. The slope of the line is μ/g_c.

A Bingham plastic, on the other hand, is a straight line that does not pass through the origin. It is represented by the equation

$$\tau - \tau_y = \frac{\mu_p}{g_c} \left(-\frac{dv_r}{dr} \right) \quad (10.11)$$

Where: τ_y = the intercept
 μ_p = plastic viscosity, a value found from the line slope

A Bingham plastic will not flow until the shear stress exceeds a value represented by τ_y , known as the *yield point*. Drilling fluids consisting of colloidal clay particles suspended in liquid approximate Bingham plastic behavior.

Pseudoplastic and *dilatant* fluids possess no yield point and exhibit nonlinear behavior. In addition, there are fluids where behavior is time dependent. A *thixotropic* fluid will produce a double curve. When increasing shear rates are imposed it will break down with time. If shear rate is then reduced no further, breakdown occurs. After exposure to high shear rates, this type of liquid will regain its original consistency after a period of time. This property is useful in special applications.

A *rheopectic* liquid is one where at constant shear rate the shear stress increases with time, the opposite of a thixotropic liquid.

It is common to characterize non-Newtonian liquids by an empirical power law formula

$$\tau = K \left(-\frac{dv_r}{dr} \right)^n \quad (10.12)$$

Where: K = an overall measure of the liquid viciousness
 = μ/g_c for Newtonian fluid
 n = 1.0 for Newtonian fluid
 = 0.0 – 1.0 for pseudoplastic fluids
 = >1.0 for dilatant fluids

Equation 10.12 is a straight line on a log-log plot. The deviation of "n" from 1.0 (in either direction) is a measure of the degree of non-Newtonian behavior.

The detailed calculation of the behavior of non-Newtonian liquids is beyond the scope of this book, but more details are available in Reference 10.3 and many references covering rheology of liquids.

Most oils may be treated as Newtonian liquids, particularly those with a relative density below 0.9. However, some emulsions of oil and water may exhibit non-Newtonian characteristics.

NEWTONIAN LIQUID FLOW

The Moody chart (Figure 10.1) is a very sound correlation if one can estimate roughness. Unfortunately, one never knows what it is at any one time, and it changes with time. Initially the pipe may contain mill scale from its manufacture. After use this mill scale may be removed by liquid action and the pipe wall becomes smoother. On continued use erosion/corrosion, scale formation and the like may increase roughness.

Figure 10.3 is based on field tests. The smooth curve is for 8 in. pipe or larger, or smooth tubing. As diameter increases, roughness has less effect.

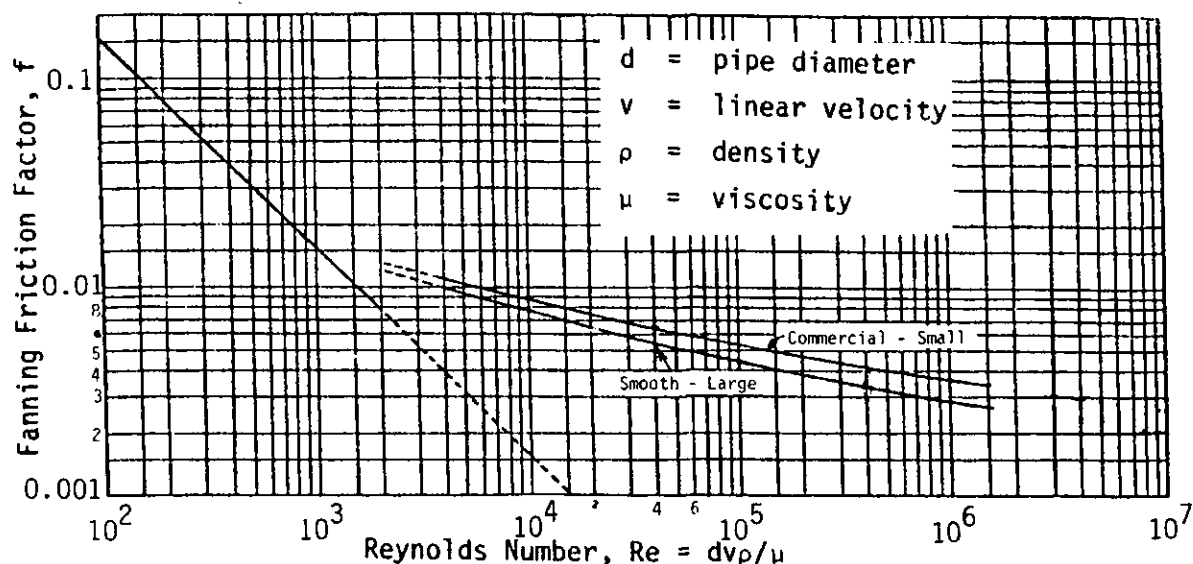


Figure 10.3 Simplified Correlation for Fanning Friction Factor

The equations of the three lines in Figure 10.3 are represented closely by the following equations.

$$\text{Re} < 2000, \quad f = 16/\text{Re} \quad (10.13)$$

$$\text{Re} > 4000, \quad f = 0.042/\text{Re}^{0.194} \quad \text{large pipe} > 20 \text{ cm [8 in.]} \quad (10.14)$$

$$\text{Re} > 4000, \quad f = 0.042/\text{Re}^{0.172} \quad \text{small pipe} \leq 20 \text{ cm [8 in.]} \quad (10.15)$$

Figures 10.1 and 10.3 have served for years as a basis in evaluation of friction factors. In the world of computers, equations are more convenient. Churchill and Usagi have proposed a single equation to replace the f vs. Re plots shown.

$$f = 2 \left[\left(\frac{8}{\text{Re}} \right)^{12} + \frac{1}{(C_1 + C_2)^{1.5}} \right]^{0.0833} \quad (10.16)$$

$$\begin{aligned} \text{Where: } C_1 &= \{2.457 \ln[1/((7/\text{Re})^{0.9} + 0.27 (\epsilon/d))]\}^{16} \\ C_2 &= (37530/\text{Re})^{16} \end{aligned}$$

For the flow of an incompressible liquid in a section of line of fixed diameter where no work is done, Equation 10.6 may be written

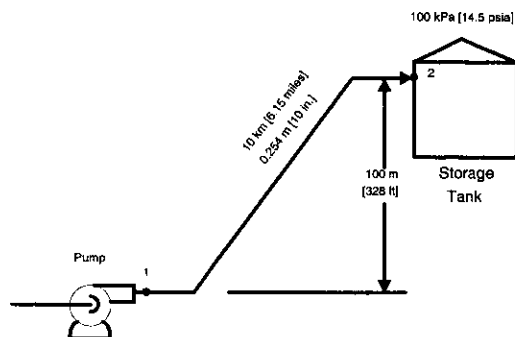
$$\begin{aligned} \Delta P &= (P_2 - P_1) = \left(-\frac{2 f L v^2 \rho}{g_c d} \right) - \rho \Delta X (g/g_c) \\ &= (-\Delta P_f) - \rho \Delta X (g/g_c) \end{aligned} \quad (10.17)$$

It often is more convenient to express flow rate in volumetric or mass terms rather than velocity. The v^2 in Equation 10.17 may be substituted for as follows:

$$v^2 = \frac{q^2}{(0.785)^2 d^4} = \frac{m^2}{\rho^2 (0.785)^2 d^4} \quad (10.18)$$

As with Reynolds number, the substitution of these terms in the Fanning friction factor equation to obtain ΔP_f enables one to write Equation 10.17 in terms of volumetric or mass flow.

Example 10.1: A 0.254 m [10 in.] crude oil line handles 6400 m³ [40 200 bbl] per day of a 0.83 Rel. ρ oil. The oil viscosity is 9.5 cp. The flow line is 10 km [6.15 miles] long and discharges to a 100 kPa [14.5 psi] storage tank. The tank battery is 100 m [328 ft] higher than the pump discharge. Determine the necessary pump discharge pressure.



(This type of problem is frequently found in new design work. No attempt is made to calculate losses caused by pipe fittings. Such losses are relevant, however, in real world situations and should be calculated. Normally they are added as an equivalent length to the actual length of the pipeline. See later text. The ΔP_f , then, should usually include any such fitting friction losses.)

Metric:

$$d = 0.254 \text{ m}$$

$$v = \frac{6400}{(86400)(0.785)(0.254)^2} = 1.46 \text{ m/s}$$

$$\mu = (9.5)(0.001) = 0.0095 \text{ kg/m}\cdot\text{s}$$

$$\rho = (0.83)(1000) = 830 \text{ kg/m}^3$$

$$\frac{d v \rho}{\mu} = \frac{(0.254)(1.46)(830)}{0.0095} = 3.2 \times 10^4$$

From Figure 10.3, $f = 0.0058$

$$\Delta P_f = \frac{(2)(0.0058)(10\,000)(1.46)^2(830)}{(1.0)(0.254)} = 8.08 \times 10^5 \text{ N/m}^2 = 8.08 \times 10^5 \text{ Pa}$$

$$\begin{aligned} \rho \Delta X \text{ g/g}_c &= (830)(100)(9.81) = 814\,230 \text{ N/m}^2 \\ (100\,000 - P_1) + 814\,230 &= -8.08 \times 10^5 \\ P_1 &= 1.72 \times 10^6 \text{ Pa} = 1.72 \text{ MPa(a)} \end{aligned}$$

English:

$$\Delta P_f = \frac{(2)(0.0058)(6.15)(5280)(4.79)^2(51.8)}{(32.2)(0.833)} = 16\,690 \text{ lbf/ft}^2$$

$$\rho \Delta X = (51.8)(328) = 16\,990 \text{ lbf/ft}^2$$

$$(2088 - P_1) + 16\,990 = -16\,690, \quad P_1 = 35\,678 \text{ lbf/ft}^2$$

In Example 10.1 ΔX is positive since the liquid is going uphill. If the liquid had been flowing downhill, ΔX would have been a negative number. Also, in this example no effort was made to specify exact pipe ID, the nominal diameter was used. In reality, pipe ID varies from the values shown in tables, since some pipe specifications as much as 12.5% deviation in pipe wall thickness. As a blunt, practical matter, the use of exact diameters causes no real increased precision in predicting long line performance.

Calculation of Pipe Diameter

Situations often occur where it is desired to calculate the line diameter (d) needed for a given flow rate and pressure drop. The friction factor, f , and Reynolds number, Re , are a function of diameter and velocity terms. Therefore, the use of the correlation of f plotted against Re involves a trial-and-error solution if diameter is desired.

A direct solution for diameter is possible by algebraic manipulation of the basic equations.

Solving for " d " yields

$$d = 1.265 q^{0.4} \left[\frac{f L \rho}{(\Delta P_f) g_c} \right]^{0.2} \quad (10.19)$$

The friction factor in Equation 10.19 can be found by many methods. It is convenient to use Equations 10.14 and 10.15. Substituting these equations for " f " into Equation 10.19 gives

$$\text{For small pipe,} \quad d = 0.649 q^{0.379} \rho^{0.172} \mu^{0.036} \left[\frac{L}{(\Delta P_f) g_c} \right]^{0.207} \quad (10.20)$$

$$\text{For large pipe,} \quad d = 0.647 q^{0.376} \rho^{0.168} \mu^{0.041} \left[\frac{L}{(\Delta P_f) g_c} \right]^{0.208} \quad (10.21)$$

where units are defined as those listed on page 4.

Example 10.2: Determine the diameter for an oil flow rate of $0.0416 \text{ m}^3/\text{s}$ [$1.46 \text{ ft}^3/\text{s}$], relative density of 0.79, a total pressure drop allowance of 500 kPa [72.5 psi] over 20 km [12.4 miles], and a viscosity of 10 cp. Assume large pipe ($d \geq 10$ inches):

$$\begin{aligned} \text{Metric:} \quad d &= 0.6474 (0.0416)^{0.376} [0.79 (1000)]^{0.168} [10 (0.001)]^{0.041} \left[\frac{20\,000}{(500\,000) 1.0} \right]^{0.208} \\ &= 0.2547 \text{ m} \end{aligned}$$

$$\begin{aligned} \text{English:} \quad d &= 0.6474 (1.46)^{0.376} [0.79 (62.4)]^{0.168} [10(0.000672)]^{0.041} \left[\frac{(12.42)(5280)}{(72.5)(144)(32.2)} \right]^{0.208} \\ &= 0.834 \text{ ft} = (10 \text{ inches}) \end{aligned}$$

Calculation of Line Capacity

The calculation of the capacity of a line with a given pressure drop is trial-and-error when using a plot of friction factor vs. Re . However, a direct solution is possible. Starting with the same basic equations as in the calculation of the diameter, eliminating q instead of d , and using Equations 10.14 and 10.15 for " f ,"

$$\text{For small pipe,} \quad q = \frac{3.127 d^{2.64}}{\rho^{0.453} \mu^{0.094}} \left[\frac{(\Delta P_f) g_c}{L} \right]^{0.547} \quad (10.22)$$

$$\text{For large pipe,} \quad q = \frac{3.180 d^{2.661}}{\rho^{0.446} \mu^{0.107}} \left[\frac{(\Delta P_f) g_c}{L} \right]^{0.553} \quad (10.23)$$

Calculation of Pressure Drops

A similar derivation approach for the remaining variable normally required in field problems, ΔP_f , yields the following equations for normal pipe friction.

$$\text{Small Pipe, } \frac{\Delta P_f}{L} = \frac{q^{1.828} \rho^{0.828} \mu^{0.172}}{8.038 g_c d^{4.828}} \quad (10.24)$$

$$\text{Large Pipe, } \frac{\Delta P_f}{L} = \frac{q^{1.806} \rho^{0.806} \mu^{0.194}}{8.081 g_c d^{4.806}} \quad (10.25)$$

Example 10.3: $q = 0.0416 \text{ m}^3/\text{s}$ [1.46 ft³/sec] $\mu = 10 \text{ cp}$
 $\rho = 790 \text{ kg/m}^3$ [49.3 lb_m/ft³] $L = 20 \text{ km}$ [12.42 miles]
 $d = 0.254 \text{ m}$ [0.833 ft]

Assume fittings increase L by 5% and elevation changes insignificant

$$\text{Metric: } \Delta P_f = \frac{(0.0416)^{1.806} (790)^{0.806} [10 (0.001)]^{0.194} (20\,000)(1.05)}{8.081 (1.0)(0.254)^{4.806}} = 535\,418 \text{ Pa}$$

$$\text{English: } \Delta P_f = \frac{(1.46)^{1.806} (49.3)^{0.806} [10 (0.000\,672)]^{0.194} (12.42)(5280)(1.05)}{8.081 (32.17)(0.833)^{4.806}} \\ = 11\,071 \text{ psf} = 76.9 \text{ psi}$$

Valves and Fittings

Equations 10.20-10.25 adequately describe flow of Newtonian fluids in straight runs of commercial steel pipe under conditions of fully turbulent flow. The pressure drop through valves and fittings must be addressed separately. The most common method used to account for this additional pressure drop is the use of equivalent lengths (L_e). Each fitting (e.g. valve, tee, el, etc.) is assigned an L_e based on its size and geometry. These are documented extensively in fluid flow literature. Reference 10.4 provides an excellent review of the theory of this approach and Reference 10.5 tabulates L_e 's for the most common valves and fittings encountered in the hydrocarbon industry. Figure 10.4 is a handy nomograph for estimating L_e for various types of fittings.

Once the L_e of a fitting has been determined it is simply added to the length of the straight pipe sections. The pressure drop calculation is then performed on the total pipe length including the L_e additions.

In the planning mode, how does one estimate the pressure drop through a pipe section before the actual pipe layout has been determined. References 10.16 and 10.7 detail a method whereby pressure drops can be estimated before detailed drawings have been completed. This is useful for predicting pumping and compression requirements. The correlation accounts for a typical number of pipe fittings and valves installed in a piping system.

$$L_e/L = 1 + (0.347 d^{1/2} + 0.216) F_c \quad (10.26)$$

Where: L_e/L = total equivalent lengths per length of straight pipe
 d = nominal pipe diameter, inches
 F_c = complexity factor (see table at right)

	F_c
Very complex piping manifolds	4
Manifold type piping	2
Normal piping	1
Long reasonably straight runs	0.5
Utility supply lines, outside battery limits	0.25

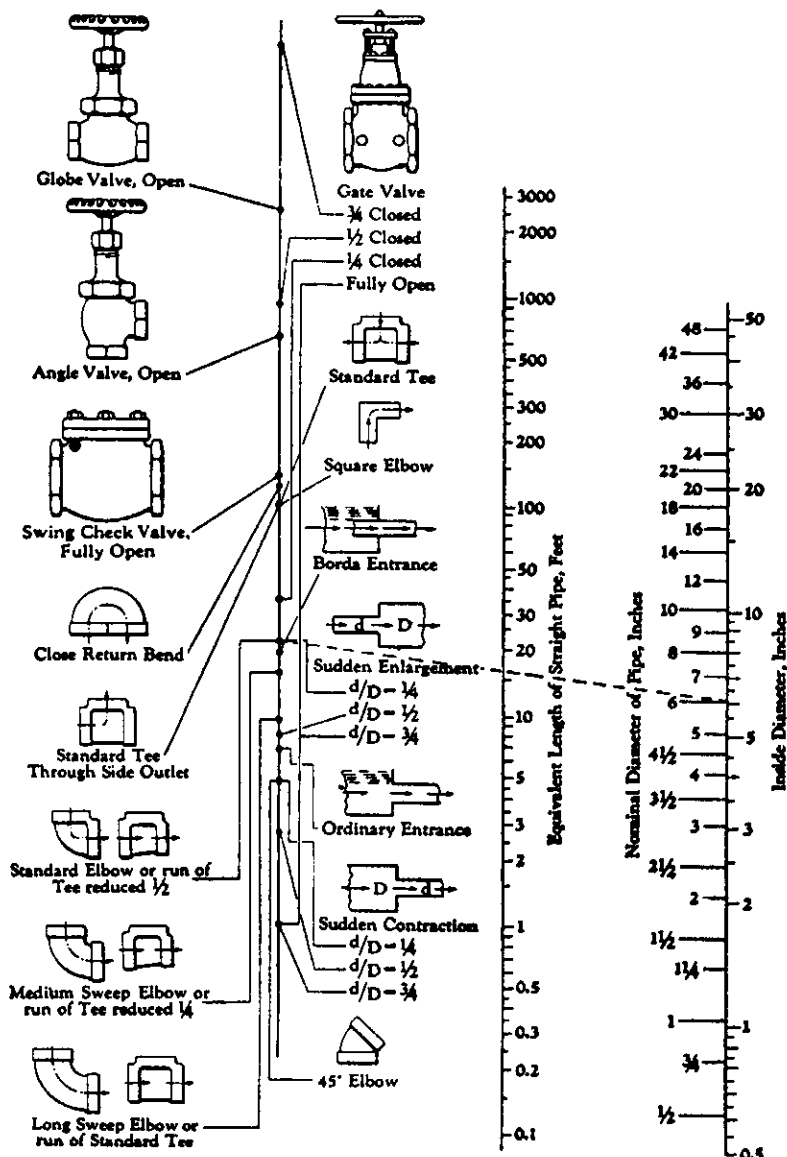
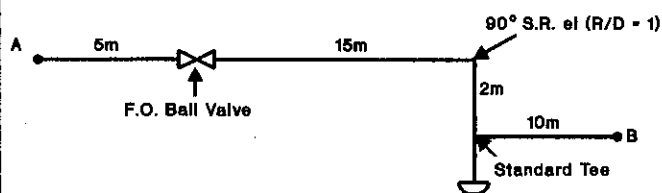


Figure 10.4 Resistance of Valves and Fittings to Flow of Fluids

Example 10.4: Calculate the effective length of piping in the system shown below. Pipe size is 0.254 m [10 in.].



From Figure 10.4

10" F.O. Ball Valve	$L_e = 2.1 \text{ m [7 ft]}$
10" 90° S.R. el	$L_e = 5.5 \text{ m [18 ft]}$
10" tee (branch flow)	$L_e = 14.3 \text{ m [47 ft]}$

For ΔP calculation, the effective length of the system

$$= 5 + 2.1 + 15 + 5.5 + 2 + 14.3 + 10 = 53.9 \text{ m [177 ft]}$$

Economic Pipe Diameter

Equations 10.20-10.25 simply provide the relationship between flow rate, pressure drop and pipe size. They do not provide line sizing criteria. In sizing a line one is always faced with a compromise of two factors. For a given flow rate of a given fluid, as pipe diameter is increased piping cost increases. But, pressure loss decreases, which reduces potential pumping or compression costs.

The economic diameter will be the one which makes the sum of amortized capital cost plus operating cost a minimum. This total cost can be per unit time or per unit of production. One can correlate this total cost (per unit time or production) versus diameter to determine a minimum.

Equation 10.27 may be used to estimate this economic diameter. It is derived from cost data and must be considered an approximation only.

$$d = \frac{A m^{0.45}}{\rho^{0.31}} \quad (10.27)$$

Where: d = economic diameter
 m = mass flow rate, in thousands of
 ρ = fluid density
 A = constant

Metric	English
m	in.
kg/h	lb/hr
kg/m ³	lb/ft ³
0.189	2.2

Equation 10.27 is derived primarily for relatively short lengths of piping in processing and production installations where pressure drop is not particularly large. It is applicable to both vapor and liquid lines. Remember also that the most economic diameter may not be the one you should choose. There are mechanical considerations which are of concern.

Many companies provide pipe sizing guidelines in terms of velocity or pressure drop per unit length. The following guidelines have proven useful for preliminary sizing of process piping.

Liquid Lines	Recommended Velocity	
	m/s	ft/s
Noncorrosive liquid flow	2-3	7-10
Corrosive liquid flow (sour condensate, glycol, amine)	0.7-1.0	2-3
Centrifugal pump suction	0.7-1.0	2-3
Reciprocating (plunger) pump suction	0.3	1
Vapor Lines	Recommended $\Delta P/L$	
	Pa/m	psi/100 ft
Natural gas: 0-700 kPa(g) [0-100 psig]	110-230	0.5-1.0
700-3500 kPa(g) [100-500 psig]	230-450	1.0-2.0
3500-14 000 kPa(g) [500-2000 psig]	450-1130	2.0-5.0

Maximum Velocity

Some company specifications limit *maximum continuous* liquid velocity to

$$v_{\max} = \frac{K A}{\rho^{0.5}} \quad (10.28)$$

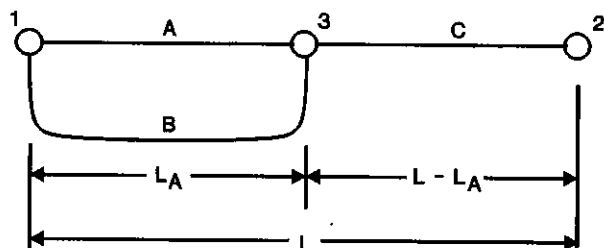
Where: v = velocity
 ρ = density
 A = conversion constant
 K =

Metric	English
m/s	ft/sec
kg/m ³	lb/ft ³
1.23	1.0
100	100

API 14E recommends three values of K; 100 for normal design calculation, 125 for intermittent service, 150 for the "never exceed" value. These values are, in general, based on erosion corrosion limitations in bends, elbows, and fittings. Some companies have found considerably higher values of K are acceptable for straight piping with "clean" fluids.

DESIGNING LOOP SYSTEMS

The need often arises for increasing the flow rate per unit pressure drop in a system. This is normally encountered where it is desired to increase the capacity or where the flow rate is to remain the same with a lower pressure. The former is a common need in expanding systems, while the latter usually stems from older systems wherein the pipe or pumping equipment has to be derated for pressure.



The usual and most economical solution to the problem is to place one or more lines parallel to the original, either partially or throughout the entire length. The figure at left shows a schematic view of a simple loop system that may be used to illustrate the principles involved.

The loop and the line it parallels are usually equal in length throughout their common length, although conditions sometimes dictate a loop of different length.

Loop Capacity

It is often desirable to calculate the capacity of an existing looped system. The most straightforward method of analyzing this problem is to convert the loop into a single line having the length of the looped section with a diameter (d_e) which gives the line the same capacity as the looped section. An alternative approach is to fix the diameter of the line and calculate the equivalent length L_e of the line so that the line capacity is equal to that of the loop.

This method can be derived from Equation 10.22 or 10.23. Equation 10.23 will be used as an example:

$$q_A = \frac{3.180 d_A^{2.661}}{\rho^{0.446} \mu^{0.107}} \left[\frac{\Delta P_f}{L_A} g_c \right]^{0.553}$$

A similar equation can be written for q_B . Now envision a line with a diameter d_e and length L_e which has the same capacity as the looped section ($q_A + q_B$).

$$q_A + q_B = \frac{3.180 d_e^{2.661}}{\rho^{0.446} \mu^{0.107}} \left[\frac{\Delta P_f}{L} g_c \right]^{0.553} \quad (10.29)$$

Remember that $(\Delta P_f)_A = (\Delta P_f)_B = (\Delta P_f)_e$. Substituting Equation 10.23 into 10.29 for q_A and q_B and canceling like terms gives

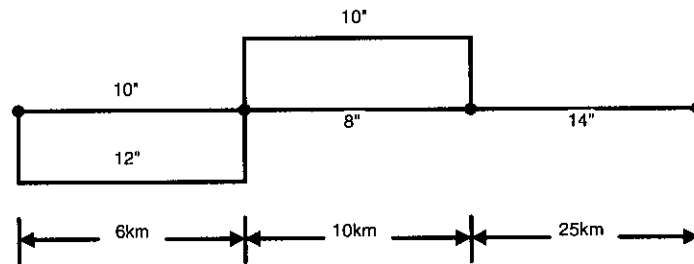
$$\frac{d_e^{2.661}}{L_e^{0.553}} = \frac{d_A^{2.661}}{L_A^{0.553}} + \frac{d_B^{2.661}}{L_B^{0.553}} \quad (10.30)$$

By an identical approach, for small pipe,

$$\frac{d_e^{2.641}}{L_e^{0.547}} = \frac{d_A^{2.641}}{L_A^{0.547}} + \frac{d_B^{2.641}}{L_B^{0.547}} \quad (10.31)$$

Equation 10.30 can be used to convert complex looped systems into simple systems for capacity calculations.

Example 10.5: Calculate the capacity of the following system handling a 32°API gravity crude oil with a viscosity of 3 cp. Allowable pressure drop is 1000 kPa [145 psi].



Convert the first looped section into an equivalent length, L_e , of 14 in. pipe –

$$\frac{14^{2.661}}{L_e^{0.553}} = \frac{10^{2.661}}{6^{0.553}} + \frac{12^{2.661}}{6^{0.553}}, \quad \text{solve for } L_e \quad L_e = 5.3 \text{ km}$$

Similarly for the second looped section –

$$\frac{14^{2.661}}{L_e^{0.553}} = \frac{10^{2.661}}{10^{0.553}} + \frac{8^{2.661}}{10^{0.553}}, \quad \text{solve for } L_e \quad L_e = 22.8 \text{ km}$$

Total effective length of system = 5.3 + 22.8 + 25 = 53.1 km

Calculate q from Equation 10.23 –

$$\gamma = 141.5 / (131.5 + 32) = 0.865 \quad \rho = 865 \text{ kg/m}^3 \text{ [54 lbm/ft}^3\text{]}$$

$$\mu = 0.003 \text{ kg/m}\cdot\text{s} \text{ [2.02} \times 10^{-3} \text{ lbm/ft}\cdot\text{sec}]$$

$$\Delta P/L = 1000000 / 53100 = 18.83 \text{ Pa/m [633 (lbf/ft}^2\text{)/ft]}$$

$$d = 0.356 \text{ m [1.17 ft]}$$

$$q = \frac{(3.180)(0.356)^{2.661}}{(865)^{0.446} (0.003)^{0.107}} [18.83]^{0.553} = 0.094 \text{ m}^3/\text{s [51 100 bpd]}$$

Loop Length

Another common application of the loop calculation is to estimate the length of loop required to increase the capacity of the system a given amount. This calculation can be made by establishing a pressure balance for the system

$$\left(\frac{\Delta P_f}{L} \right)_A (L_A) + \left(\frac{\Delta P_f}{L} \right)_C (L - L_A) = (\Delta P_f)_{\text{total}} \quad (10.32)$$

If we define $X = L_A/L$ (fraction of the system looped) we can write the equation

$$\left(\frac{\Delta P_f}{L}\right)_A (X) + \left(\frac{\Delta P_f}{L}\right)_C (1 - X) = \left(\frac{\Delta P_f}{L}\right)_{\text{total}} \quad (10.33)$$

Equations 10.24 and 10.25 can be used to calculate $(\Delta P_f/L)_A$ and $(\Delta P_f/L)_C$. The right hand term $(\Delta P_f/L)_{\text{total}}$ is fixed by allowable pressure drop. Equation 10.33 can be solved for X .

Direct substitution of Equations 10.24 and 10.25 into Equation 10.33 yields the equations shown in Table 10.1.

TABLE 10.1
Equations for Lines in Parallel

	Lines in Parallel	
	Large Pipe	Small Pipe
Equivalent Diameter	$\frac{d_e^{2.661}}{L_e^{0.554}} = \frac{d_A^{2.661}}{L_A^{0.554}} + \frac{d_B^{2.661}}{L_B^{0.554}}$	$\frac{d_e^{2.641}}{L_e^{0.547}} = \frac{d_A^{2.641}}{L_A^{0.547}} + \frac{d_B^{2.641}}{L_B^{0.547}}$
Looping Requirements*	$X = \frac{1 - (q/q_1)^{1.806}}{1 - \left[\frac{d_A^{2.661}}{d_A^{2.661} + d_B^{2.661}} \right]^{1.806}}$	$X = \frac{1 - (q/q_1)^{1.828}}{1 - \left[\frac{d_A^{2.641}}{d_A^{2.641} + d_B^{2.641}} \right]^{1.828}}$
Entire Line Looped*	$\frac{q_1}{q} = \left[1 + \left(\frac{d_B}{d_A} \right)^{2.661} \right]$	$\frac{q_1}{q} = \left[1 + \left(\frac{d_B}{d_A} \right)^{2.641} \right]$
Diameter of Original and Parallel Lines the Same*	$X = 1.40 \left[1 - \left(\frac{q}{q_1} \right)^{1.806} \right]$	$X = 1.392 \left[1 - \left(\frac{q}{q_1} \right)^{1.828} \right]$
*Assumes that the length of all lines in the loop are the same length.		

For Lines in Parallel:

Where:

- d_e = diameter of a single line that is equivalent to a group of parallel lines
- d_A = diameter of the original line before looping
- d_B = diameter of a single loop line (or equivalent diameter of a group of loop lines)
- L_e = length of equivalent single line corresponding to d_e
- L_A = length of loop line d_A
- L_B = length of loop line d_B
- q = original flow rate before line is looped
- q_1 = flow rate after looping line
- X = fraction of length of original line that is looped

Example 10.6: It is desired to increase the capacity of an existing pipeline system by 60%. The diameter of the existing line is 12 inches. What percentage of the pipeline system must be looped if the diameter of the loop is 14 inches?

$$X = \frac{1 - (1.0/1.6)^{1.806}}{1 - \left[\frac{12^{2.661}}{12^{2.661} + 14^{2.661}} \right]^{1.806}} = \underline{0.706} = \underline{70.6\%}$$

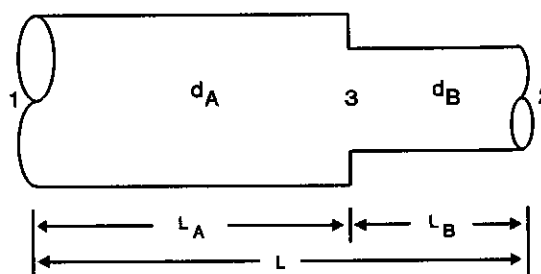
Complex Liquid Gathering Systems

The situation often arises when it is convenient to use several diameters of pipe in series. Fluid principles also may be applied in this case after proper analysis of the conditions present.

In the sketch at right, L and d represent the length and diameter, respectively, of two different size lines joined in series. Therefore:

$$q_A = q_B = q_{\text{total}} \quad \text{and} \quad \Delta P_{fB} + \Delta P_{fA} = \Delta P_{f \text{ total}}$$

However, no one equation may be written between points (1) and (2) because of the varying diameter. It should be possible to write an equation for section 1-3 and one for 3-2, solve each for q_L , and permit one to solve for the remaining unknown P_3 .



A more flexible and convenient solution is afforded by determining a single-diameter line that is equivalent to the system shown. An equation in the form of 10.24 or 10.25 may be written for one section. Then the equation is written for a hypothetical section having the same diameter as the other section but of such length that the ΔP_f is the same as that first written. Then:

$$\Delta P_f = \frac{C L_e}{d_A^{5-n}} = \Delta P_{fB} = C \left(\frac{L_B}{d_B^{5-n}} \right) \quad (10.34)$$

where L_e is defined as the length of line of diameter d_A that will give the same friction loss as length L_B of line d_B . C is defined as all the terms of equation 10.24 or 10.25 not related to length or diameter. If the physical properties of the fluid remain essentially unchanged throughout the line the C terms are equal. If the constants are canceled and Equation 10.34 rearranged:

$$L_e = L_B \left(\frac{d_A}{d_B} \right)^{5-n} \quad (10.35)$$

where n is the exponent in Reynolds number when the friction factor is expressed in the form of Equation 10.14 or 10.15.

If d_e is defined as the diameter of line of length L_A that will give the same friction loss as L_B feet of line d_B , Equation 10.34 may be rearranged to yield:

$$d_e = d_B \left(\frac{L_A}{L_B} \right)^{\frac{1}{(5-n)}} \quad (10.35a)$$

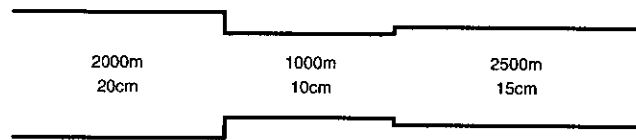
Equation 10.35 is particularly useful, for the system shown then may be regarded as consisting of $L_e + L_A$ feet of line of diameter d_A . Once this equivalence is established, the problem may be solved in the conventional manner.

Equation 10.33 is summarized for large pipe and small pipe in Table 10.2.

TABLE 10.2
Equations for Lines in Series

	Lines in Series	
	Large Pipe	Small Pipe
Equivalent Diameter	$d_e = d_B \left(\frac{L_A}{L_B} \right)^{0.208}$	$d_e = d_B \left(\frac{L_A}{L_B} \right)^{0.207}$
Equivalent Length	$L_e = L_B \left(\frac{d_A}{d_B} \right)^{4.806}$	$L_e = L_B \left(\frac{d_A}{d_B} \right)^{4.828}$

Example 10.7: Express the equivalent length of the following system in terms of 10 cm pipe, assuming that the nominal diameter is the actual inside diameter.

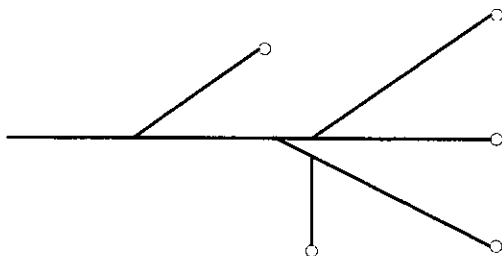


From Equation 10.35,

$$L_{10} = (2000)(10/20)^{4.828} + (2500)(10/15)^{4.828} = 423 \text{ m}$$

$$\text{Total length} = 423 + 1000 = 1423 \text{ m}$$

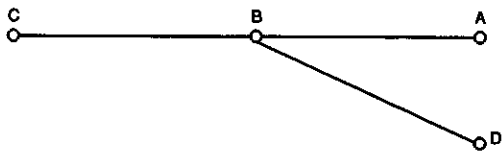
Therefore, the system shown may be considered as equivalent to one consisting of 1423 m of 10 cm pipe. Capacity or pressure-drop calculations then may be made on the latter system using the correlations previously presented.



Most gathering systems contain one or more branch lines. If the system is very complex, one has a large number of calculations to make. As a practical matter, these will be made on a computer.

This essentially is a trial-and-error approach, although some short cuts are possible. What one will do in principle is to use the fundamental equations discussed previously and combine these with the concepts of series and parallel flow.

The total system is made up of a series of branch lines coming together at a common point. In the sketch



$$q_{CB} = q_{AB} + q_{BD}$$

and P_B is common to all three lines. In the usual case, the flow rate from each well or production facility is estimated from reservoir analysis and the design of the down-hole equipment. The problem is to establish line sizes and line arrangement to optimize initial cost and pressure loss.

The proper approach is to plan the total gathering system at the time field development commences. Spacing and deliverability can be estimated within reason to set up an expected grid. Some modification will

be needed usually, but the system will be more efficient than one which is expanded in a rather haphazard manner. Too often the resultant is an operating nightmare and is quite inefficient.

The procedure used for system design may be varied. One approach may be illustrated using the simple sketch shown previously.

1. Establish q_{AB} , q_{BD} , P_A and P_D . This fixes q_{CB} .
2. Establish a reasonable pressure for Point C.
3. Guess a pressure for Point B, between pressures established at A, C and D. One method is to prorate on distance.
4. Calculate d for each line, using methods outlined previously.
5. Repeat the calculation for various combinations of pressures, flow rates and line configurations until a grid is found that is satisfactory.

For preliminary studies we sometimes take a short cut by assuming a line velocity of 2–3 m/s [6–10 ft/sec] for each well flow rate to establish a diameter for its line. With these d 's and map lengths a tentative grid may be drawn. ΔP is found for each leg and one can "work through" the grid to find the pressure distribution. In most cases, this simplified approach will check more complex methods rather well.

In most liquid grids the oil properties are relatively constant, as are wellhead pressures. If this is the case

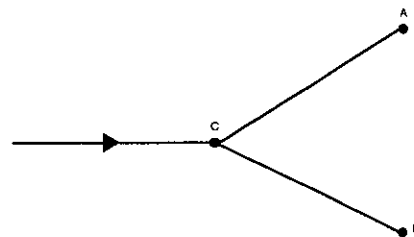
$$\Delta P_f \propto (d, v, L)$$

One can set up a series of relative numbers for calculation of the system. If P_A and P_D are at the same pressure and at about the same elevation, lines A-B and B-D behave as loop lines and an equivalent diameter may be found.

We can use the principles to algebraically develop many working equations that describe the specific system being studied.

Flow Splitting

A common problem also encountered is where the flow from a single line is split into two or more parts, as in the figure at the right. In the rare case where the pressures and elevations at A and B are equal, this may be handled as a loop problem – in all others the calculation of flow is theoretically a trial-and-error process. However, this tedious approach usually may be eliminated by the following method.



1. Assume a pressure at Point C that is high.
2. Calculate ΔP_f for lines CA and CB.
3. Calculate q_L for both lines.
4. Knowing the total flow rate at C and the relative q_L 's from above, calculate the actual flow rate for each line.
5. Using the actual flow rates, calculate ΔP_f for lines CA and CB using a correlation of f vs. Re .
6. Calculate pressure at C for both lines.
7. If the two pressures at C check within a few percent, the actual flow rates calculated in (4) above may be used. If not, a new value must be assumed for the pressure at C and the calculation repeated.

In most instances the above process is only necessary one time, particularly if the pressure at C is assumed high enough. Often it will work even if the assumed pressure and that calculated in Step 6 differ widely. As a rule-of-thumb, the assumed pressure should be about 15 times the maximum elevation change, expressed in pressure terms.

GAS FLOW

Principles

The fundamental thermodynamic equations used for liquid flow also may be used for gases. In order to conveniently handle Equation 10.6, several assumptions are often made:

$$\int V dP + \frac{g}{g_c} \Delta X + \frac{(\Delta v)^2}{2 g_c} = - \frac{2 f L v^2}{g_c d} - W$$

1. That no external work is done by or on the system, i.e., $W = 0$.
2. That the flow is isothermal.
3. That the changes in elevation on a long pipeline are negligible, i.e., $X = 0$.
4. The flow is steady state.

These three assumptions allow Equation 10.6 to be rewritten as:

$$\int V dP = \frac{-2 f L v^2}{g_c d} \quad (10.36)$$

To evaluate the integral term the following assumption is made

$$v = \frac{n R T z_m}{P} \quad (10.37)$$

where z_m is not a function of pressure over the integration range.

Substitution of Equation 10.37 in Equation 10.36 and integration gives the "Basic Equation" shown below:

$$q_{sc} = K \left[\frac{T_{sc}}{P_{sc}} \right] \left[\frac{(P_1^2 - P_2^2) d^5}{f \gamma L T_m z_m} \right]^{0.5}$$

All of the above assumptions are usually satisfactory on the typical long pipeline. However, those equations derived on this basis usually contain an efficiency factor E to correct for these inaccuracies. (See Table 10.3) The assumption of isothermal flow has little effect on the final accuracy, which may be proven by assuming adiabatic conditions, the opposite extreme.

The use of a mean compressibility factor z_m is convenient. It may be determined at the average pressure P_m , which in turn is found from the equation:

$$P_m = 2/3 \left[\frac{P_1^3 - P_2^3}{P_1^2 - P_2^2} \right] = 2/3 \left[(P_1 + P_2) - \left(\frac{P_1 P_2}{P_1 + P_2} \right) \right] \quad (10.38)$$

where P_1 and P_2 are the inlet and outlet absolute pressures, respectively. The mean temperature can be found from the equation:

$$T_m = \left[\frac{T_1 - T_2}{\ln \left(\frac{T_1 - T_g}{T_2 - T_g} \right)} \right] + T_g \quad (10.39)$$

Where: T_g = temperature of pipe surroundings

TABLE 10.3
Summary of Gas Flow Equations

	Metric	English
Basic, Eq. 10.40 $q_{sc} = K \left[\frac{T_{sc}}{P_{sc}} \right]^{1.000} \left[\frac{(P_1^2 - P_2^2) d^5}{f \gamma L T_m z_m} \right]^{0.5} (E)$	$K = 5.62 \times 10^5$	$K = 38.774$
Weymouth, Eq. 10.41 $q_{sc} = K \left[\frac{T_{sc}}{P_{sc}} \right]^{1.000} \left[\frac{(P_1^2 - P_2^2) d^{5.333}}{\gamma L T_m z_m} \right]^{0.5} (E)$	$K = 1.162 \times 10^7$ $f = \frac{0.00235}{d^{0.33}}$	$K = 433.49$ $f = \frac{0.008}{d^{0.33}}$
Panhandle A, Eq. 10.42 $q_{sc} = K \left[\frac{T_{sc}}{P_{sc}} \right]^{1.0788} \left[\frac{(P_1^2 - P_2^2) d^{4.854}}{\gamma^{0.8541} L T_m z_m} \right]^{0.5394} (E)$	$K = 1.198 \times 10^7$ $f = \frac{0.0189}{(q\gamma/d)^{0.1461}}$	$K = 435.87$ $f = \frac{0.0192}{(q\gamma/d)^{0.1461}}$
Panhandle B, Eq. 10.43 $q_{sc} = K \left[\frac{T_{sc}}{P_{sc}} \right]^{1.02} \left[\frac{(P_1^2 - P_2^2) d^{4.961}}{\gamma^{0.961} L T_m z_m} \right]^{0.51} (E)$	$K = 1.264 \times 10^7$ $f = \frac{0.0057}{(q\gamma/d)^{0.03922}}$	$K = 737$ $f = \frac{0.00359}{(q\gamma/d)^{0.03922}}$
AGA, Eq. 10.44 $q_{sc} = K \left[\frac{T_{sc}}{P_{sc}} \right]^{1.000} \left[\frac{(P_1^2 - P_2^2) d^5}{\gamma L T_m z_m} \right]^{0.5} (F_t)$	$K = 5.622 \times 10^5$	$K = 38.774$
	Partially Turbulent	Fully Turbulent
	$F_t = F_f \sqrt{1/f_{SPL}}$ $= F_f \left[4 \log \left(\frac{Re}{\sqrt{1/f}} \right) - 0.6 \right]$	$F_t = 4 \log (3.74 d/\epsilon)$

Where:

q_{sc} = gas rate at T_{sc} , P_{sc}
 P = absolute pressure
 P_{sc} = pressure, standard conditions
 T_m = mean absolute temperature of line
 T_{sc} = temperature, standard conditions
 T_g = ground temperature
 d = inside diameter of pipe
 ϵ = absolute roughness
 L = pipe length
 μ = viscosity
 γ = gas relative density
 z_m = mean compressibility factor
 f = Fanning friction factor
 E = pipeline efficiency
 Re = Reynolds number
 F_t = transmission factor ($\sqrt{1/f}$)
 F_f = drag factor

Metric	English
m^3/d	scf/d
kPa	psia
kPa	psia
K	°R
K	°R
K	°R
m	in.
m	in.
m	mile
Pa·s	lb/ft-sec
—	—
—	—
—	—
—	—
—	—
—	—
—	—

Standard Equations

Table 10.3 summarizes several standard equations derived from the basic flow equation based on the above mean conditions. Equation 10.40 is the basic equation containing a friction factor "f." The Weymouth and Panhandle A and B equations are merely the basic Equation 10.40, using the friction factor correlations shown. The Weymouth equation assumes "f" depends only on pipe diameter; the Panhandle equations use different correlations for "f" as a function of gas flow rate, relative density and pipe diameter. The AGA equation uses the "f" values shown in Table 10.3.

The use of a *transmission factor* $(1/f)^{0.5}$ is common in gas transmission. It is a function of Re. Figure 10.5 shows the basic relationship for the flow conditions shown. The Weymouth equation is a horizontal line for fully turbulent flow. The smooth pipe law (no roughness) and Panhandle A parallel the AGA equation for partially turbulent flow. The AGA equation for fully turbulent flow is a series of horizontal lines depending on pipe roughness (ϵ). This will vary, but a value of 46 μm [1800 $\mu\text{in.}$] is suitable for typical steel pipe.

Although Figure 10.5 shows a sharp change between partially turbulent and turbulent flow, there really is a transition zone. If one wishes to use an AGA equation, use Equation 10.45 to determine which one.

$$\text{Re} = 20.91 \left(\frac{d}{\epsilon} \right) \left[\log \left(\frac{3.7 d}{\epsilon} \right) \right] \quad (10.45)$$

If the actual Re is less than that calculated from Equation 10.45, use the partially turbulent form; if greater, use the turbulent flow equation. For most large diameter natural gas transmission lines operating dry in a non-corrosive environment, $\epsilon = 15 - 33 \mu\text{m}$ [600 - 1300 $\mu\text{in.}$] with 19 μm [750 $\mu\text{in.}$] a reasonable average.

In Example 10.8 the range of answers obtained applies only to this example, but is indicative of the spread usually obtained. In the final analysis, the usefulness of a given equation will depend on experience with it in varying circumstances. The Weymouth equation is used quite often on gas gathering system design since it maximizes pipe diameter needs for a given flow and pressure drop.

Example 10.8: Determine flows using the Basic, Weymouth, Panhandle A and B, and AGA fully turbulent equations (Table 10.3) from the following information.

		Metric	English
Where:	P_1 = absolute pressure	3448 kPa	500 psia
	P_2 = absolute pressure	3103 kPa	450 psia
	d = inside diameter of pipe	0.3048 m	12 in.
	γ = gas relative density	0.7	0.7
	L = pipe length	16 100 m	10 miles
	T_m = mean absolute temperature of line	300 K	541 °R
	z_m = mean compressibility factor	0.96	0.96
	μ = viscosity	0.01 cp	0.01 cp
	E = pipeline efficiency	0.92	0.92
Solutions:	q_{sc} = gas rate at T_{sc} , P_{sc}	10^6 std m^3/d	MMscfd
	Basic (using Figure 10.1)	1.3	45
	Weymouth	1.1	39
	Panhandle A	1.3	44
	Panhandle B	1.4	50
	AGA (maximum)	1.3	47

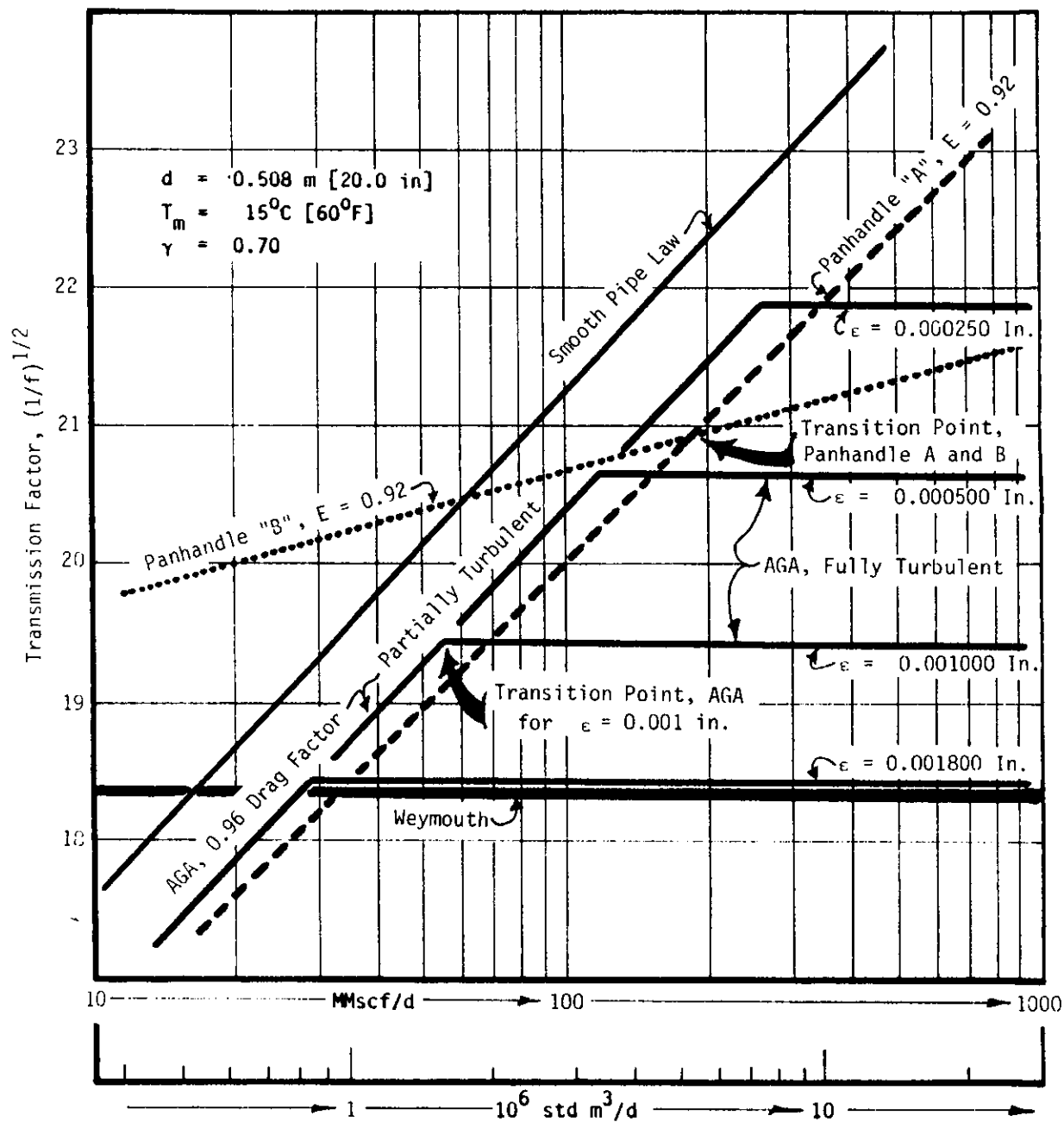


Figure 10.5 Transmission Factor Comparison

Maximum Velocity

The maximum allowable velocity in a gas line is governed by noise, pressure loss and surge considerations. Equation 10.46 is a guideline for establishing any maximum velocity limits. Design velocity always should be less than this maximum velocity.

$$v = \frac{A}{(\rho)^{0.5}} \quad (10.46)$$

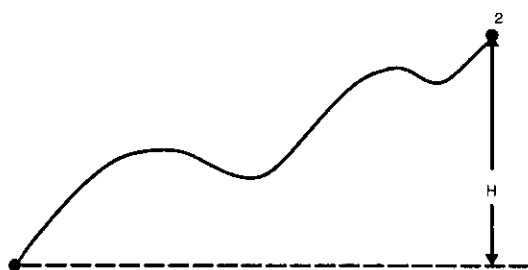
Where: v = velocity
 ρ = density
 A = constant

Metric	English
m/s	ft/sec
kg/m ³	lb/ft ³
146	120

At a pressure of about 7.0 MPa [1000 psia] Equation 10.46 will show a maximum velocity of about 17-18 m/s. At about 14.0 MPa the maximum velocity will be about two-thirds of this amount. See page 2211 for additional comments.

Static Pressure (Head) in Flow Lines

All equations like those shown in Table 10.3 are based on the assumption that the line is horizontal; i.e., there is no potential energy change affecting P_1 and P_2 . In actual practice the line may be going uphill or downhill.



For single phase flow some correction must be made for the relative elevation between Points (1) and (2). Before using these equations some correction must be made for P_1 or P_2 , to convert the ΔP to what it would have been if the line were horizontal.

Since the gas is compressible and temperature varies, many models may be used. The simplest equation uses an average temperature and compressibility, ignores any kinetic energy changes and assumes a constant friction factor. For these assumptions, from the basic energy equations, one may derive the equation

$$\frac{H \gamma}{A T_m z_m} = \ln \left(\frac{P_A}{P_B} \right) \quad (10.47)$$

Where: H = head
 γ = gas relative density
 T_m = mean gas temperature
 z_m = mean gas compressibility
 P_A = pressure at bottom of static column
 P_B = pressure at top of static column
 A = constant

Metric	English
m	ft
—	—
K	°R
—	—
MPa	psia
MPa	psia
29.28	53.34

Equation 10.47 also can be written in the form

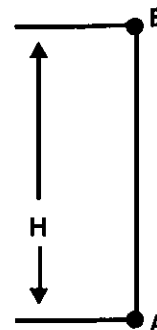
$$P_A = P_B e^s \quad (10.48)$$

Where: $s = \gamma/AT_m z_m$ and $e = 2.718$

There are two ways in which one may handle this correction for head, depending on which is more convenient for the calculation involved. Both require knowledge about the value of H , the relative elevation between (1) and (2).

What we are doing is converting the line from the actual profile shown previously to that shown at right. If line pressure P_1 is fixed, the pressure at (2) for a given circumstance depends on whether the line is horizontal, goes uphill or downhill.

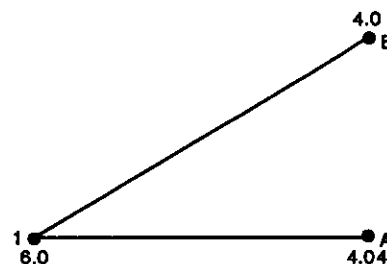
We use the subscripts A and B to show the pressure change due to head only. Do not confuse these with measured (or specified) pressures P_1 and P_2 . In the uphill case, P_2 for use in the horizontal flow equations will be greater than the actual P_2 . In Equation 10.48, actual P_2 will be inserted as P_B . The value of P_A solved for will be the new P_2 for use in the horizontal flow equations. For downhill flow the effective horizontal P_2 will be less than the actual P_2 and the process with Equation 10.48 is reversed.



One simple approach for a nonhorizontal line is to convert the actual P_2 (if known) to the equivalent P_2 for a horizontal line which can then be used in the type of equations shown in Table 10.3.

Example 10.9: A gas line runs uphill. The following data apply:

$$\begin{aligned}
 H &= 100 \text{ m}, \quad T_m = 300 \text{ K}, \quad z_m = 0.90, \quad \gamma = 0.70, \\
 P_2 &= 4.0 \text{ MPa}, \quad P_1 = 6.0 \text{ MPa} \\
 s &= (100)(0.70)/(29.28)(300)(0.9) = 0.00885 \\
 P_A &= (4.0)e^{0.00885} = 4.04 \text{ MPa (which is equivalent } P_2 \text{ for} \\
 &\quad \text{horizontal flow)}
 \end{aligned}$$



If the line had been horizontal, the total pressure drop would have been less and P_2 would have been higher, or P_2 in the above example would have been 4.04 MPa. This value for P_2 would then be used in an equation of the type shown in Table 10.3. If the line goes downhill (the net H is minus), the equivalent horizontal P_2 would be less than the actual P_2 . The basic rule – use as P_2 in the pipeline equations the value that should occur if the line is horizontal to relate q , d and pressure drop as shown by the equation chosen.

Also remember that length L should be established from the line profile and not merely the geographical distance between (1) and (2).

The alternative is to incorporate the pressure and length corrections directly into the flow equations. Probably, the method shown is easier and clearer.

For most actual gas pipelines the correction for head and length is very small. If in doubt, make it. The value of the calculation depends upon its use in the decision process.

SINGLE-PHASE VERTICAL GAS FLOW

The correction for potential energy (head) in the previous section is the simplest possible approach but is nevertheless suitable for situations where the magnitude of H is not large. It is thus suitable for the calculation of static pressure in shallow gas wells.

I cannot recommend the equations of Table 10.3 for flow in a wellbore. Each of these equations is based on the premise that the friction factor is primarily a function of diameter, a reasonable conclusion in relatively large diameter piping. Most tubing and casing possess small diameters compared to pipelines. Thus, other equations are often used for the application.

Static Bottom Hole Pressure

Equations 10.47 and 10.48 serve as a simple model for estimating bottom hole static pressure for a gas column. P_B would be the wellhead pressure and P_A the bottom hole pressure, under static conditions.

This is a trial-and-error calculation since z_a cannot be calculated until P_A is known. The first step is to assume a value of P_A to find z_a . Solve Equation 10.47 or 10.48 for P_A using this z_a . Use this new P_A to find a new z_a . After 3-4 trials the P_A assumed will converge on the P_A calculated.

What about the first assumption for P_A ? A reasonable first guess is to assume a 600 kPa [90 psi] per 300 m [1000 ft] of depth.

An average temperature will have to be assumed for use with this simple approach. The use of any average introduces error. One way to compromise this is to subdivide total depth into several sections for calculation purposes. The P_A for the top section is the P_B for the next section, and so on down. In this approach, averages are not taken over as wide an interval of P and T and are thus more representative of the gradient.

There are two ways to find average quantities. One is to average P and T and then find the quantities at these average conditions. The second is to calculate the value of the desired quantities at each known P and T and then average each quantity. The effect of the averaging method on results is negligible in most instances.

Cullender and Smith^(10.8), Poettmann^(10.9) and others have proposed solutions for static bottom-hole pressure that are represented by Equation 10.49

$$\int_{P_1}^{P_2} \frac{z dP}{P} = \frac{\gamma H}{AT_m} = \int_{P_{r1}}^{P_{r2}} \frac{z dP_r}{P_r} = \int_{0.2}^{P_{r2}} \frac{z dP_r}{P_r} - \int_{0.2}^{P_{r1}} \frac{z dP_r}{P_r} \quad (10.49)$$

The left-hand terms of Equation 10.49 are equivalent to Equation 10.47. The value of the integrals on the right can be found from Table 10.4.

TABLE 10.4
Integrals of $(z/P_r)dP_r$ versus P_r

Pseudo-Reduced Pressure, P_r	Pseudo-Reduced Temperature, T_r										
	1.05	1.10	1.20	1.30	1.40	1.50	1.6	1.8	2.0	2.4	3.0
0.2	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00
0.4	0.62	0.62	0.63	0.63	0.63	0.64	0.64	0.64	0.64	0.64	0.64
0.6	0.96	0.97	1.00	1.02	1.04	1.05	1.05	1.05	1.05	1.05	1.05
0.8	1.18	1.21	1.26	1.30	1.32	1.34	1.35	1.36	1.36	1.36	1.36
1.0	1.33	1.38	1.46	1.50	1.53	1.55	1.57	1.59	1.60	1.61	1.61
1.2	1.43	1.50	1.60	1.65	1.68	1.71	1.74	1.76	1.78	1.80	1.81
1.4	1.49	1.59	1.71	1.77	1.81	1.85	1.88	1.91	1.94	1.95	1.98
1.6	1.53	1.65	1.80	1.88	1.92	1.96	1.99	2.04	2.06	2.08	2.11
1.8	1.56	1.69	1.87	1.96	2.01	2.06	2.09	2.14	2.17	2.20	2.23
2.0	1.59	1.73	1.92	2.03	2.09	2.14	2.18	2.23	2.27	2.30	2.33
2.5	1.66	1.82	2.04	2.18	2.26	2.32	2.36	2.44	2.48	2.51	2.56
3.0	1.73	1.89	2.14	2.29	2.40	2.47	2.51	2.60	2.64	2.69	2.74
3.5	1.80	1.96	2.23	2.39	2.52	2.58	2.64	2.74	2.79	2.85	2.90
4.0	1.86	2.03	2.31	2.47	2.62	2.69	2.75	2.86	2.92	2.98	3.04
4.5	1.93	2.10	2.38	2.56	2.70	2.78	2.85	2.97	3.03	3.10	3.16
5.0	2.00	2.17	2.45	2.63	2.78	2.86	2.94	3.07	3.13	3.21	3.27
5.5	2.07	2.24	2.52	2.70	2.86	2.94	3.02	3.15	3.22	3.30	3.37
6.0	2.13	2.30	2.59	2.77	2.93	3.02	3.10	3.24	3.30	3.39	3.46
7.0	2.26	2.43	2.71	2.90	3.06	3.15	3.24	3.39	3.46	3.56	3.62
8.0	2.39	2.56	2.84	3.03	3.18	3.29	3.37	3.52	3.60	3.70	3.77
9.0	2.51	2.68	2.96	3.15	3.31	3.41	3.49	3.64	3.73	3.82	3.90
10.0	2.63	2.80	3.08	3.26	3.42	3.52	3.61	3.76	3.84	3.94	4.02

Example 10.10: A 0.74 relative density natural gas is produced from a reservoir with a subsurface depth of 1768 m [5800 ft]. The shut-in surface pressure of this well is 12.4 MPa [1800 psia]. From well logs the reservoir temperature is 53°C [128°F]; the wellhead temperature is 20°C [68°F]. Estimate the bottom hole pressure.

$$P_c = 4.58 \text{ MPa [665 psia]} \quad \text{and} \quad T_c = 214 \text{ K [385°R]}$$

English:

$$T_m = (68 + 128)/2 = 98^\circ\text{F}$$

$$(Hy)/(AT_m) = (5800)(0.74)/(53.34)(98 + 460) = 0.144$$

$$P_r = 1800/665 = 2.71, \quad T_r = 558/385 = 1.45$$

From Table 10.4 (by interpolation) the value of the integral from 0.2 to $P_{r1} = 2.35$

$$\text{From Equation 10.49,} \quad \int_{0.2}^{P_{r2}} (z/P_r) dP_r - 2.35 = 0.144 \quad ,$$

$$\int_{0.2}^{P_{r2}} (z/P_r) dP_r = 2.49$$

By interpolation from Table 10.4, $P_{r2} = 3.3$

$$P_2 = (665)(3.3) = 2195 \text{ psia}$$

Relatively simple methods are available for calculating vertical gas flow. References 10.8-10.10 summarize these. One approach is to use Equation 10.40 (Table 10.3) with the value for "f" found from the equation following (from Cullender and Binckley).

$$f = A \left[\frac{\mu^{0.065}}{q^{0.065} d^{0.058} \gamma^{0.065}} \right] \quad (10.50)$$

Where:

μ = gas viscosity
 q = gas flow rate
 d = pipe diameter
 γ = gas specific gravity
 A = constant

Metric	English
cp	cp
10^6 std m/d	MMscf/d
m	ft
—	—
0.005 715	0.007 73

An excellent review of these methods is provided in a series by Aziz.^(10.10)

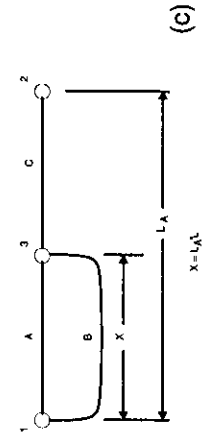
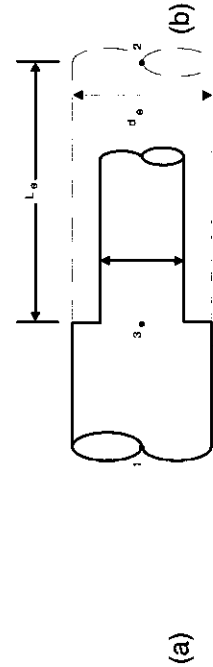
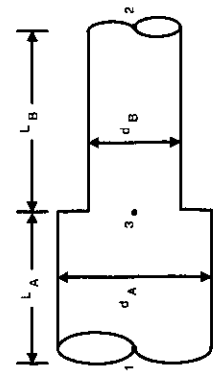
Complex Gas Flow Systems

The principles outlined for liquid flow also apply to steady-state gas flow. Table 10.5 summarizes the resulting equations. The sketches below the table outline the nomenclature used. The subscript (e) denotes a single line equivalent in length or diameter to another line in series or looped lines.

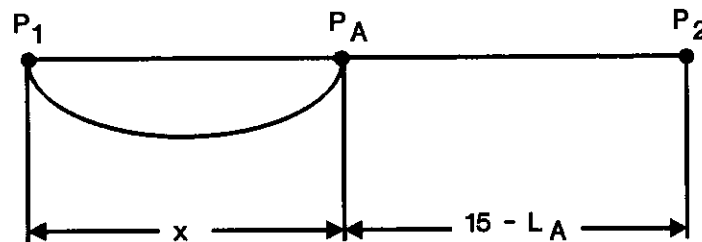
Examples 10.11-10.12 will serve to illustrate the application of the gas flow equations. For simplicity, the Weymouth equation, uncorrected for gas compressibility, will be used as an illustration of the principles involved.

TABLE 10.5
Equations for Complex Gas Flows

	Basic	Weymouth	Panhandle
For Two Lines in Series			
Equivalent Diameter, d_e	$d_B \left(\frac{L_A}{L_B} \right)^{1/5} \left(\frac{f_A}{f_B} \right)^{1/5}$	$d_B \left(\frac{L_A}{L_B} \right)^{3/16}$	$d_B \left(\frac{L_A}{L_B} \right)^{0.2060}$
Equivalent Diameter, L_e	$L_B \left(\frac{d_A}{d_B} \right)^5 \left(\frac{f_B}{f_A} \right)$	$L_B \left(\frac{d_A}{d_B} \right)^{16/3}$	$L_B \left(\frac{d_A}{d_B} \right)^{4.854}$
For Two Lines in Parallel			
Equivalent Diameter or Length - Loops d_e or L_e	$\frac{d_e^{2.5}}{(f_e L_e)^{0.5}} = \frac{d_A^{2.5}}{(f_A L_A)^{0.5}} + \frac{d_B^{2.5}}{(f_B L_B)^{0.5}}$	$\frac{d_e^{8/3}}{L_e^{1/2}} = \frac{d_A^{8/3}}{L_A^{1/2}} + \frac{d_B^{8/3}}{L_B^{1/2}}$	$\frac{d_e^{2.618}}{L_e^{0.5394}} = \frac{d_A^{2.618}}{L_A^{0.5394}} + \frac{d_B^{2.618}}{L_B^{0.5394}}$
Loops - Diameters and Flows Vary X = Fraction Looped $d_R = \left(\frac{1/f_B}{1/f_A} \right)^{0.5} \left(\frac{d_B}{d_A} \right)^{5/2}$	$X = \frac{1 - \left(\frac{q}{q_1} \right)^{0.5}}{1 - \frac{1}{(1 + d_R)^2}}$	$X = \frac{1 - \left(\frac{q}{q_1} \right)^2}{1 - \left[\frac{d_A^{8/3}}{d_A^{8/3} + d_B^{8/3}} \right]^2}$	$X = \frac{1 - \left(\frac{q}{q_1} \right)^{1.86}}{1 - \left[\frac{d_A^{2.618}}{d_A^{2.618} + d_B^{2.618}} \right]^{1.86}}$
Entire Line Looped		$\frac{q_1}{q} = 1 + \left(\frac{d_B}{d_A} \right)^{8/3}$	$\frac{q_1}{q} = 1 + \left(\frac{d_B}{d_A} \right)^{2.618}$
Diameters of Original and Parallel Lines are the Same X = Fraction Looped	$X = 4/3 \left[1 - \left(\frac{q}{q_1} \right)^2 \right]$	$X = 4/3 \left[1 - \left(\frac{q}{q_1} \right)^2 \right]$	$X = 4/3 \left[1 - \left(\frac{q}{q_1} \right)^{1.86} \right]$



Example 10.11: A portion of a large gas-gathering system consists of a 15.41 cm [6.067 in.] line 15 km [9.4 miles] that is handling 206×10^3 std m^3/d [7.6 MMscfd] with an average Rel. ρ of 0.64. The pressure at the upstream end of this section is 2.58 MPa gauge [375 psig] and the average delivery pressure is 2.07 MPa gauge [300 psig]. The average temperature is 23°C [73°F].



Due to new well completion, it is desired to increase the capacity of this line 20% by looping with additional 15 cm line. What length is required? Let L_A represent the length of the loop section.

Metric Solution

$$\text{New flow rate } q_1 = 1.2 (206 \times 10^3) = 247.2 \times 10^3 \text{ std m}^3$$

The loop may be represented by a single line d_e .

$$d_e = [(15.41)^{8/3} + (15.41)^{8/3}]^{3/8} = 19.98 \text{ cm} = 0.1998 \text{ m}$$

A Weymouth equation for each section may now be written.

$$247.2 \times 10^3 = 1.162 \times 10^7 \left(\frac{288}{100} \right) \left[\frac{(2680)^2 - P_A^2}{L_A (0.64)(296)} \right]^{1/2} (0.1998)^{8/3}$$

$$247.2 \times 10^3 = 1.162 \times 10^7 \left(\frac{288}{100} \right) \left[\frac{P_A^2 - (2170)^2}{(15 - L_A)(0.64)(296)} \right]^{1/2} (0.1541)^{8/3}$$

(Note: 100 kPa has been added to each gauge pressure.) Since there are two equations and two unknowns these equations may be solved algebraically. For this example $L_A = 6.0$ km.

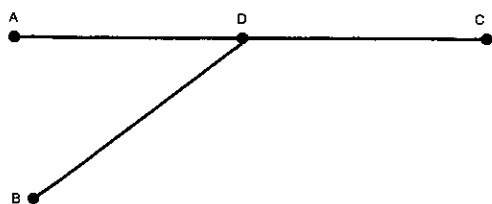
Using the equations shown in Table 10.5 for looping requirements represents an easier solution method.

$$X = \frac{1 - \left(\frac{206}{247.2} \right)^2}{1 - \left[\frac{(15.41)^{8/3}}{(15.41)^{8/3} + (15.41)^{8/3}} \right]^2} = 0.407$$

$$L_A = 0.407 (15 \text{ km}) = 6.1 \text{ km}$$

Example 10.12

The system at left is to be designed for the gathering of gas. Well "A" is a flowing gas well that will deliver 81.33×10^3 std m^3/d [3 MMscfd]. Lease "B" contains high gas-oil ratio oil wells. It is necessary to determine what compressor discharge pressure is necessary in order to deliver 47.44×10^3 std m^3/d [1.75 MMscfd] from point B. The pressure at the transmission line (point C) is 2.07 MPa gauge [300 psig]. Average flowing temperature will be $29^\circ C$ [$85^\circ F$]. The gravity of the flowing gas is 0.63 and that of the casing head gas is 0.71. The line diameters shown have been arbitrarily fixed because of pipe availability.



It is obvious that once the pressure at point D is found, P_B may be found; furthermore, the average specific gravity in line CD is estimated.

$$\frac{81.33(0.63) + 47.44(0.71)}{128.77} = 0.66$$

$$\text{Line CD} \quad 128.77 \times 10^3 = 1.162 \times 10^7 \left(\frac{288}{100} \right) \left[\frac{P_D^2 - (2170)^2}{(16100)(0.66)(302)} \right]^{1/2} (0.1521)^{8/3}$$

$$P_D = 2409 \text{ kPa}$$

$$\text{Line BD} \quad 47.44 \times 10^3 = 1.162 \times 10^7 \left(\frac{288}{100} \right) \left[\frac{P_B^2 - (2409)^2}{(4830)(0.71)(302)} \right]^{1/2} (0.1024)^{8/3}$$

$$P_B = 2490 \text{ kPa}$$

UNSTEADY STATE GAS FLOW

All of the discussion to this point is based on steady state flow. Unsteady state flow is, however, an important consideration. This cannot be analyzed thoroughly and conveniently by manual calculations but there are some simple approximations that may prove useful.

Transient Pipe Line Flow

Any time the rate of withdrawal of gas from a line differs from the input, unsteady state flow is occurring. During routine operations this often occurs because said line is used for storage to dampen out peak loads. During peak demand, output exceeds input and pressure is declining throughout the line. Between peak loads input exceeds output and the pressure builds up again.

To predict behavior of the line one must start with the basic energy equations and then develop a solvable model that will approximate the behavior of a moving pressure front with time. This is strictly a computer modeling game. Some models I see are not too applicable. Examine a model carefully to ascertain it if fits your specific system situation.

Blow Down and Purge

Occasions occur when it is necessary to blow down and purge a gas line. This is a special case of unsteady state flow.

One may estimate the blowdown time by some simple equations.^(10.11,10.12)

For the *critical flow* case (when the ratio of the higher pressure to lower pressure is larger than two):

$$t = \left(\frac{BV}{C_d A_v} \right) \left(\frac{\gamma}{zT} \right)^{0.5} \ln \left(\frac{P_1}{P_2} \right) \quad (10.51)$$

Where:

- t = blowdown time
- B = constant
- V = actual system volume
- C_d = valve discharge coefficient
- A_v = valve area
- γ = gas specific gravity
- z = average gas compressibility factor
- T = average gas temperature
- P_1 = initial system pressure
- P_2 = final system pressure

Metric	English
s	sec
0.09	5.3
m ³	ft ³
—	—
m ²	in. ²
—	—
—	—
K	°R
kPa	psia
kPa	psia

The gas temperature and compressibility factor upstream of the blowdown valve change during the blowdown period. Thermodynamically the blowdown process falls somewhere between isentropic and isenthalpic. Estimating the average temperature and compressibility factor requires some knowledge of the thermodynamic path which is not known exactly. For most blowdown cases use of the initial temperature and compressibility factor is a reasonable assumption.

Equation 10.51 was developed for critical (sonic) flow. Once the system pressure falls below about 200-300 kPa [30-50 psia] the flow is probably subcritical. Fortunately the time required to completely depressure a system to atmospheric does not usually have to be calculated. When it does, Equation 10.51 has been found to give a reasonable estimate of total blowdown time if P_2 is set to about 80% of atmospheric pressure.

Pressure Surges on Closing a Valve

When a valve is closed on a line a pressure surge occurs. This is required thermodynamically. The kinetic energy of the fluid is converted to internal energy when flow stops. A wave (surge) travels back through the line countercurrent to the fluid still flowing forward, but slowing down. Each section of the line behaves in a different manner. Since flow does not stop upstream when stoppage occurs at some point downstream, the pressure rise decreases as the wave front travels upstream.

Some of the energy is absorbed by the expansibility of the fluid and the pipe. The more energy absorbed, the less the pressure rise. Thus, compressibility of the fluid is a primary factor. Since liquids are essentially incompressible, they represent the primary problem. By the same token, high pressure gas will exhibit more pressure rise than low pressure gas.

A thorough analysis is definitely a computer solution. Reference 10.13 provides an excellent summary of the problem and some important references.

For a liquid line an approximate equation for predicting pressure rise on a valve closing is

$$\Delta P = A \left(\frac{vL}{t} \right) \quad (10.52)$$

Where:

- v = liquid velocity
- L = length of pipe preceding valve
- t = valve closing time
- ΔP = pressure rise above steady state flowing pressure
- A = constant

Metric	English
m/s	ft/sec
m	ft
s	sec
kPa	psi
5.2	0.070

Pressure Testing

Gas pressure testing of a line is a form of unsteady flow in the sense that pressure must be held for sufficient time to assure that there is no leak. The time necessary for such testing may be estimated by the equation

$$t_m = \frac{Ad^2L}{P_1} \quad (10.53)$$

Where: t_m = time necessary, minimum
 d = internal pipe diameter
 L = length of pipe section
 P_1 = initial test pressure
 A = constant

Metric	English
h	h
cm	in.
km	miles
MPa	psig
0.002	3.0

When a line has been shut in for at least the above amount of time, it may be considered "tight" if the following pressure loss has not been exceeded.

$$\Delta P_{\max} = \frac{tP_1}{Ad} \quad (10.54)$$

Where: ΔP_{\max} = acceptable pressure loss
 t = shut-in time
 d = pipe diameter
 A = constant
 P_1 = initial test pressure

Metric	English
kPa	psi
h	h
cm	in.
0.372	949
MPa	psig

TEMPERATURE CHANGES IN PIPING

Most of the equations used commonly for pressure loss calculations require the use of a single value of temperature. This, of necessity, is some kind of an average temperature. In a numerical (computer) solution one divides the line into a series of constant temperature sections; an analytical (manual) solution may treat the entire line as one section. However one does it, the temperature used should be representative of what is anticipated. The value used has a direct impact on the physical property values used.

Prediction of the temperature distribution in the flowing system may be as important as the pressure profile. It affects many design considerations surrounding the line. Crude oil pour points, gas hydrates, vapor-liquid phase behavior and water content of gas are all temperature sensitive. A reliable temperature prediction is an early step in the calculation procedure.

The temperature at any point is predictable through use of a first law thermodynamic energy balance. If potential and kinetic energy changes are ignored – and work equals zero in the pipe section involved – this first law balance for a steady state system reduces to $\Delta H = Q$.

The enthalpy of a gas increases with decreasing pressure; it is almost independent of pressure for a liquid. So, the enthalpy at any point in the line depends both on pressure at that point and the heat energy lost or gained through the pipe wall.

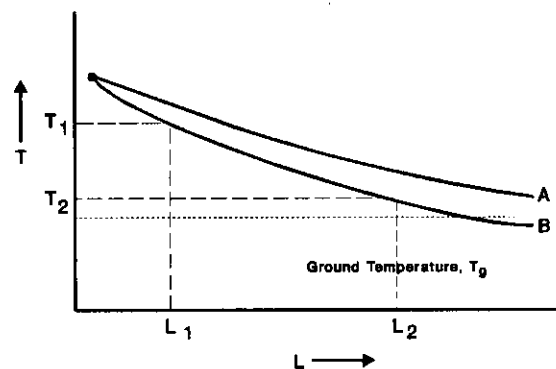
From heat transfer principles, $Q = UA\Delta t_m$. For a circular pipe this may be written as:

$$Q = (U)(\pi dL)(\Delta T_m)$$

Where: U = overall heat transfer coefficient
 $\pi = 3.1416$
 d = pipe diameter
 L = pipe section length = $L_2 - L_1$
 ΔT_m = log mean $\Delta T = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2)$
 Q = heat lost in line section of length L
 $\Delta T_1 = T_1 - T_g$
 $\Delta T_2 = T_2 - T_g$
 T_g = normal ground or water temperature at that point (far enough away from pipe to be undisturbed by it)

The general relationship is shown at right. If the fluid temperature is greater than T_g , heat transfer will be negative as shown; if it is less than T_g , the Q will be positive and the curves will approach T_g from below.

Line A is for the case where the effect of pressure on enthalpy is ignored, as per the discussion following. This is a simple heat transfer process and the fluid temperature cannot drop below T_g . Line B reflects the additional change in temperature due to the effect of pressure on enthalpy. In commercial practice this line normally would not drop over 2-3°C below T_g for ordinary pressure drop situations.



If no phase changes occur in the line section, the following equation applies.

$$\Delta h = \int_{T_1}^{T_2} mC_p dT + \int_{P_1}^{P_2} [V - T(\delta V / \delta T)_p] dP$$

This equation relates enthalpy to specific heat (C_p) and P , V and T . The second term is zero, or essentially so, for an ideal gas and a liquid. For a gas the equation shows that the enthalpy increases as the pressure declines.

If the pressure drop in the line segment (dP) is small compared to the absolute pressure, the second term of the equation is small compared to the first term. For a gas line at a pressure above 3.5 MPa [500 psi], ignoring this second term may not be important. There are thus three basic solutions for the Δh equation.

1. Use an enthalpy-PVT equation of state program.
2. Approximate the second term by assuming a series of Joule-Thomson expansions.
3. Ignore the second term completely.

Method (1) is a routine computer solution. Method (3) is an easy manual solution. The practical accuracy is as good as for (1) if a gas line is calculated in segments so that the pressure drop per segment does not exceed about 20% of the initial pressure. Method (2) is a carryover from pre-computer days as a manual method to calculate the effect of pressure on temperature (A, B). It could be programmed but is not as convenient as Method (1). Both Methods (1) and (2) require iterative solutions.

One can approximate Method (2) by translating the change of enthalpy with pressure into a temperature change per unit length. If this is done, Equation 10.55 results.

$$\ln \left[\frac{(T_1 + JL_1) - (T_g + J/a)}{(T_2 + JL_2) - (T_g + J/a)} \right] = a L \quad (10.55)$$

		Metric	English
Where:	C_p = heat capacity	kJ/kg.°C	Btu/lb-°F
	L_1 = distance from initial point	m	ft
	L_2 = distance from initial point	m	ft
	$L = L_2 - L_1$	m	ft
	T_1 = temperature at L_1	°C	°F
	T_2 = temperature at L_2	°C	°F
	γ = gas sp gr (air = 1.0)	—	—
	T_g = ground or water temperature	°C	°F
	J = Joule-Thomson coefficient	°C/m	°F/ft
	d = outside pipe diameter	m	ft
	U = overall heat transfer coefficient	kJ/h·m ² ·°C	Btu/hr·ft ² ·°F
	q = gas flow rate, std volumes	10 ³ m ³ /h	Mscf/h
	B = constant	408	24.4
	$a = dU/Bq\gamma C_p$		

Because of the Joule-Thomson effect, it is possible for the gas temperature to be less than that of the ground. This is only likely to occur with long lines possessing a large pressure drop or at regulator stations. For lean, pipeline-quality natural gases the cooling due to Joule-Thomson expansion is approximately 0.004 to 0.005 °C/kPa [0.05-0.06 °F/psi] at pipelines operating near 70 bar.

With Method (3) the quantity "J" is zero and Equation 10.56 results.

$$\ln \left(\frac{T_1 - T_g}{T_2 - T_g} \right) = \frac{d U L}{B q \gamma C_p} \quad (10.56)$$

The above equations normally are solved for T_2 . It is obtained by taking the antilog of the left-hand side. This means that any errors in the data are amplified.

Overall Heat Transfer Coefficient (U). This is the single number that represents all of the resistances in series. In a buried pipe, all of the following resistances to heat flow can occur:

1. Film coefficient between fluid and pipe wall
2. Inner pipe wall conductivity (tubing)
3. Annular space between inside and outside pipe
4. Pipe wall
5. Pipe coating
6. Insulation
7. Concrete layer or bond
8. Sand backfill
9. Native soil near pipe at temperature above normal due to heat gain from pipe

A line buried in soil normally would have resistances 1, 4, 5, 8 and 9; buried on the sea bottom 1, 4, 5 and 7 may apply. In a wellbore with fluid flowing inside the tubing 1, 2, 3, 4, 7 and 9 would be in series. The use of insulation can be justified only in very special situations.

Resistance 9 is necessary to correct for the temperature rise around the pipe above T_g , the temperature of the bulk of the ground at that depth. Temperature surveys show it is common for the ground temperature to be higher than T_g up to 5 m [15 ft] from the line. This is an effective resistance reducing heat flow.

As with a heat exchanger, the most reliable values of "U" are found by test. A study of uninsulated gathering and transmission lines shows values of U from 5-20 kJ/h-m²-K [0.25-1.0 Btu/hr-ft²-°F]. Most of the values measured are in the lower third of this range. Data on steam flood injection wells and geothermal production wells often will show higher "U" values because resistance (1) is lower.

Heat loss rates in bodies of water containing substantial currents or in tidal flats are higher because of convection losses due to water movement. A given line may cross several different heat loss environments.

The most practical approach is to recognize that a calculation based on a single value of "U" is basically invalid. Pick a likely range of values from test data to arrive at a range of answers. Then design for the worst case.

Temperature T_g . – This is not a constant quantity but depends on air temperature to some degree. At a burial depth below the "frost line" the soil temperature generally will vary from 2-16°C [35-60°F] seasonally. Maximum ground temperature will lag air temperature by a month or two. About the same temperature range will be encountered with burial in water over 30 m [100 ft] deep. In a body of water with no pronounced currents, the fluctuation will be less. Some lakes will have a bottom water variation of no more than 3°C [5°F] year round.

Temperature surveys should be a part of planning if the temperature of line contents is a design factor other than pressure loss calculations. For very preliminary considerations the minimum temperature seldom will be less than 0-1°C [32-35°F].

Because of changes in T_g , the value of "U" will vary seasonally also. In a temperate climate with discernible seasons, the "U" may be 50-60% higher in winter than in summer. In theory this should not occur, "U" being independent of T_g , but it has been noted in tests. One must recognize that the models only approximate the real world.

Above Ground Lines. – The above equations do not apply for above ground lines; no T_g is applicable. The maximum temperature of fluid in the line is the result of gain of daytime heat from the sun by radiation and the corresponding loss by convection to the air. The relative rate of these would determine the rate of heating with distance. The temperature depends on

Time of day	Wind velocity
Atmospheric conditions	Color and character of pipe surface
Air temperature (T_a)	Flow rate and properties of fluid in pipe

There are no absolutely rigorous performance prediction methods, but data are available from line tests. ^(10.35)

The maximum line temperature may be estimated by the equation

$$T = \left[\frac{R}{(\pi h_a)} \right] + T_a \quad (10.57)$$

Where: T = max temp of fluid, °F
 R = solar radiation absorbed, Btu/ft²/hr
 h_a = air film coefficient for convection, Btu/hr-ft²-°F
 π = 3.1416
 T_a = ambient air temperature, °F

Values of R and h_a may be estimated from the table below.

Pipe Surface	R	Wind Velocity, miles/h	h_a
Highly oxidized steel	300	0	2.0
Oxidized	230	2	2.6
Normal	180	5	3.5
Bright	110	10	4.5
Aluminum Paint	90	15	5.0
White Paint	75	20	5.2

Equation 10.57 shows the *maximum temperature* possible with the sun directly overhead, a clear sky and a long enough line to achieve thermal equilibrium between radiation energy gain and convection loss. In a typical case, the rise in temperature above ambient will not exceed 10-16°C [18-30°F].

The actual temperature probably will be lower than the maximum unless the exposed line is very long. Fluid properties, velocity, pipe length, season of the year (sun position) and geographical position all influence the actual radiation effect. Since not all of these are constant with time, neither is performance.

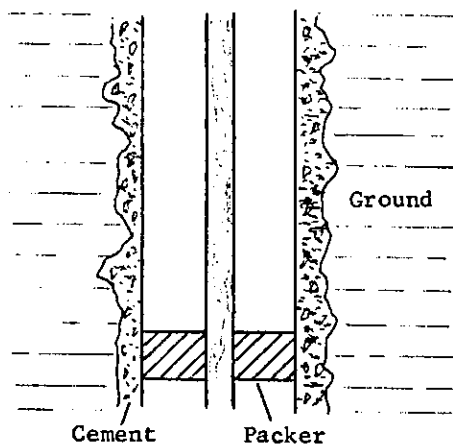
At night, even in a tropical climate, radiation losses to a clear, dark sky may be enormous. So ... rapid temperature changes can occur. The best one can hope to do is estimate the magnitude of such changes for decision purposes.

Insulation. – The purpose of insulation is to reduce the rate of gain or loss of energy, to or from a pipe, in a cost-effective manner. When one has a group of resistances in series, the one with the lowest thermal conductivity controls total heat loss.

The amount and type of insulation is governed by mechanical limitations and economics. Some very effective insulations are not suitable for buried line applications. Some of the polyurethane formulations have proven satisfactory in lines containing heated heavy oils and liquefied natural gas. Overall values of "U" of the order of 0.01 Btu/hr-ft²-°F are achievable.

Wellbore Temperatures

Prediction of wellbore temperatures parallels that for pipelines. In this case, though, ground temperatures around the pipe may vary substantially with depth. A series of models has been proposed for injection, production and geothermal wells.



The prediction of surface flowing temperatures is critical to surface facility design. None of the models is completely adequate because of the inability to define all variables quantitatively. The work of Ramey is most often used as the basic equation. Actual wellbore then is used to adjust the constants. Effective values of "U" may vary from 0.25-2.0 Btu/hr-ft²-°F. They tend to be higher than for pipelines because of higher fluid velocities and film coefficients.

The models used for calculation of well-bore temperatures are designed for computer usage. The accuracy of the output is contingent on the amount of hard data available for similar wells in the same area. In the absence of such data, the accuracy of such prediction may be no better than 15-25%. One of the common mistakes in surface facilities design is not allowing for a range of wellhead temperatures and designing for the worst condition.

Temperature depends on time. It takes finite time for the temperature profile to stabilize. This is true particularly in wellbore and gathering lines as wells are continually shut in and opened or flow rates change. With steam injection, geothermal wells and wells in permafrost or ice islands, insulation may be required to stabilize the temperature-time relationship.

References 10.14-10.24 and 10.38-10.46 address temperature performance.

GAS-LIQUID (TWO-PHASE) FLOW

The simultaneous flow of liquid and gas in a line is most important in modern operations. For many installations the use of two-phase lines is the most economical solution. A two-phase line reduces metal needed and reduces capital cost 20-25% from that of two single-phase lines. It is also practical from a mechanical standpoint.

All models are based on the same thermodynamic principles and fluid flow principles discussed previously herein. As with single-phase flow, a prediction of the "lost work" (friction drop) term is essentially empirical. Thus, any one method must be limited by the pipe sizes, fluid characteristics, flow conditions and geometry, and gas-liquid ratios used to determine a given correlation.

There is no single best correlation for universal use. When using a given model one should examine the kinds of systems on which it is based. This is why data on existing systems is vitally important. Is the model and its supporting data physically compatible with your proposed system? Like all semi-empirical equations, extrapolation beyond the range of data used to develop said equation is extremely hazardous.

For example, some correlations have been based on data from small diameter, horizontal pipes. Their application to large diameter pipelines with uphill and downhill profiles is limited. Their primary application would be in process piping. Table 10.6 from Reference 10.32 indicate the range of data taken from some of the more popular correlations.

TABLE 10.6
Experimental Information for the Two-Phase Pressure Drop Correlations

Correlation	Date	Basis	Pipe Size(s)	Fluids
Vertical Flow				
Duns & Ross	1961	Laboratory and field data	wide range	oil, gas, water
Angel-Welchon-Ross	1964	Field data	large diameter tubing and annuli	gas, water
Hagedorn & Brown	1965	Laboratory and field data	1 in. - 4 in.	oil, gas, water
Orkiszewski	1967	Review and modifications of other methods	wide range	oil, gas, water
Aziz & Govier	1972	Laboratory and field data	wide range	oil, gas, water
Beggs & Brill	1973	Laboratory data	1 in., 1.5 in.	gas, water
Gray	1974	Field data	< 3.5 in.	gas condensates
Horizontal Flow				
Lockhart-Martinelli	1949	Laboratory data	0.0586 in. - 1.1017 in.	
Eaton	1966	Laboratory and field data	2 in., 4 in.	
Dukler	1969	Data and similarity analyses	wide range	oil, gas, water
Inclined Flow				
Mukherjee-Brill	1983	Laboratory data	1.5 in.	kerosene, lube oil, gas

In the real world there are three systems of basic concern – a pipeline with uphill and downhill components, vertical pipes in wellbores and risers, and combinations of the two. Correlations from these kinds of systems are of primary importance.

The discussion herein is merely an overview of the principles involved, to provide basic understanding of the principles. Chapter 8 of Reference 10.1 provides a more detailed discussion and references.

A large amount of work has been done in transparent piping so that behavior could be photographed. A number of flow regimes have been observed which help our understanding of the problem.

Figure 10.6 shows various forms of flow noted. Different words might be used to describe each type, and a different number of types could be used, but the seven regimes shown cover the range observed.

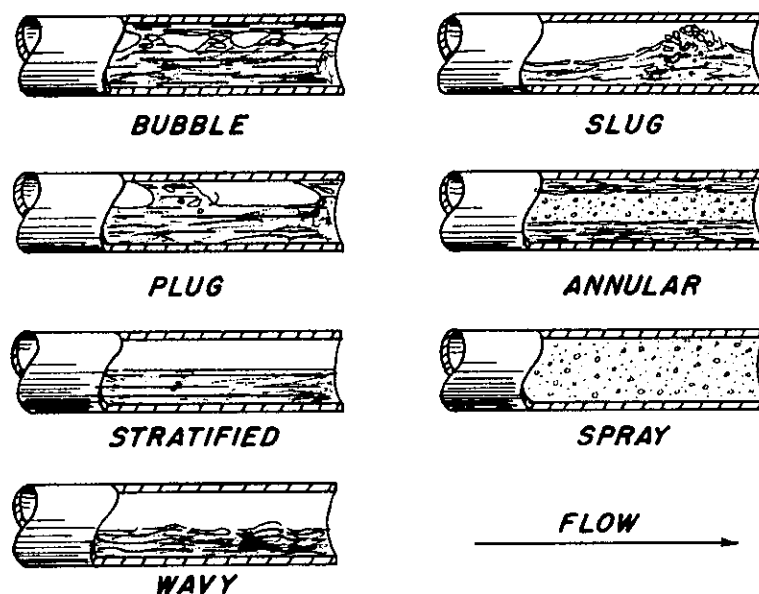


Figure 10.6 Various Two-Phase Horizontal Flow Regimes

The basic behavior of these two-phase systems depends on whether or not both the liquid and gas phases are present as a *continuous* phase; i.e., a continuous channel for flow exists for the phase. In bubble flow one has a series of bubbles of gas in the liquid. Gas is not a continuous phase. At the other end of the scale one sees spray or mist flow – small particles of liquid suspended in the gas. Liquid is now the discontinuous phase, gas the continuous phase.

In both cases the discontinuous phase may be considered to be merely altering the flow behavior of the continuous phase. One very simple approach is merely to modify the properties and/or friction factor of the continuous phase and then treat the system as single-phase flow. In bubble flow a liquid, single-phase calculation would be made. In spray or mist flow a gas correlation is used after this same correction.

This often is done by default when correlating data on existing systems. Most crude oil lines contain some gas and most gas lines contain at least entrained liquid. This is why we need separators and scrubbers.

In some of the regimes shown in Figure 10.6 both phases are continuous. This is a more complicated system because of the interactions between discrete phases. The gas moves more rapidly than the liquid. There is a shear force at the phase boundary. The character of flow depends on such variables as phase densities and viscosities, the velocity of each phase, gas-liquid ratio and properties like surface tension.

In a given length of line several flow regimes might occur because of varying forces and gas-liquid ratios. The latter changes as liquid condenses from gas or gas is formed from liquid, as dictated by phase behavior. Consideration of variables like these is a necessary part of developing a correlation.

Figure 10.7 shows four basic regimes that occur in vertical flow. A fifth regime, not shown, may be called froth flow. As noted later, this occurs at high liquid throughputs when the gas bubbles in the liquid are dispersed in an ever-increasing number of small bubbles as gas throughput increases.

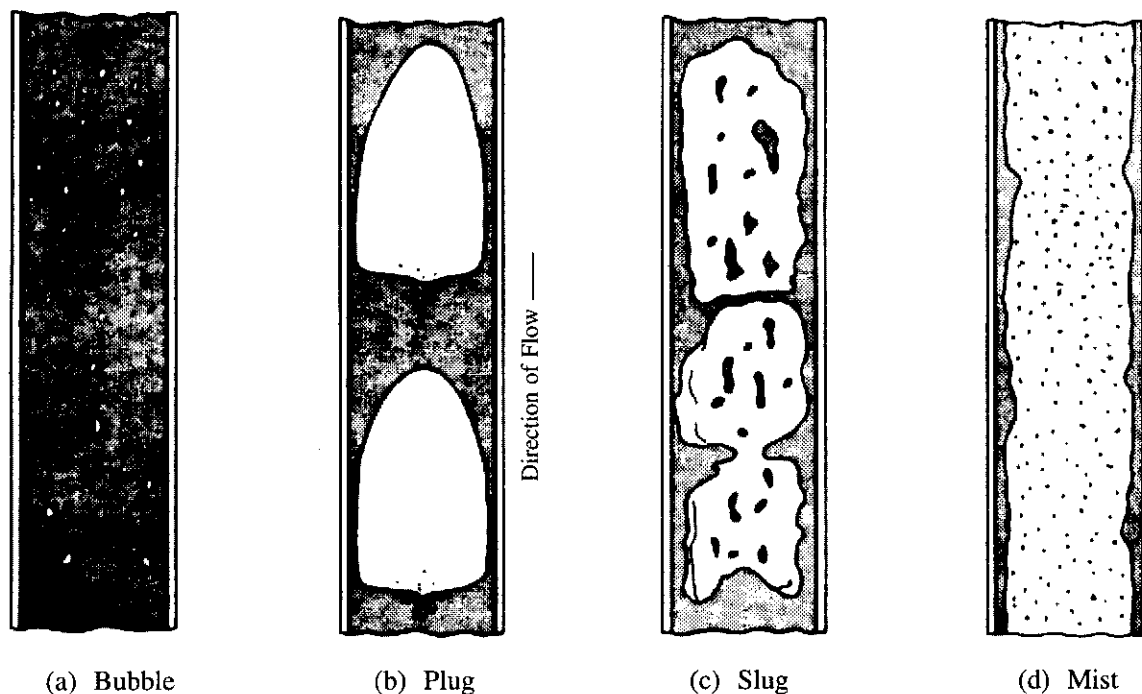


Figure 10.7 Vertical Two-Phase Flow Regimes

In vertical flow the force of gravity opposes the dynamic forces (instead of being at right angles to it). This results in *slippage*. Consequently, vertical flow exhibits some different characteristics than horizontal flow and may be more complicated.

Consider friction loss for a moment. The friction factor is a measure of all irreversible energy changes occurring in the pipe. In single-phase flow the friction factor primarily is a representation of the loss between fluid and the pipe wall and is correlated as a function of fluid properties, fluid velocity, pipe diameter and pipe roughness.

In two-phase flow there are two phases, with different properties, interacting with the pipe wall as well as with each other. A friction factor, therefore, depends on more, and complex, variables. The difference between correlations depends to a large extent on how these variables are treated in order to fit the available data.

Because of the complex forces involved in determination of an effective friction factor, several dimensionless groups in addition to Reynolds number have proven useful.

Froude Number

This dimensionless group represents the ratio of inertial force to gravity force. It can be used to characterize flow whenever gravity force influences fluid flow.

$$Fr = \frac{v}{(gd)^{0.5}} \quad \text{or} \quad Fr = \frac{v^2}{gd}$$

The form at left is the basic equation. The right-hand expression is oftentimes more convenient to use and is often referred to as a Froude No. even though it is actually $(Fr)^2$.

Weber Number

The dimensionless group is defined as the ratio of inertial force to surface force. It is equal to twice the ratio of kinetic energy to surface energy of a given volume of fluid. It may be written as

$$We = \frac{v}{(\sigma/\rho d)^{0.5}} \quad \text{or} \quad We = \frac{v^2 \rho d}{\sigma}$$

Once again the right-hand form often is used because of its convenience. This dimensionless group is a logical correlation parameter where one phase is discontinuous because interfacial tension (σ) affects drop size.

Euler Number

This dimensionless group is the ratio of applied pressure force to inertial force. It is a suitable correlating function in a flow system where pressure force is a controlling force, particularly if one of the fluids is compressible. It is written as

$$Eu = \frac{v}{(2 \Delta P/\rho)^{0.5}} \quad \text{or} \quad Eu = \frac{\rho v^2}{2 \Delta P}$$

In the above three dimensionless groups the nomenclature is as follows:

v = effective fluid velocity	ρ = fluid density
g = acceleration of gravity	σ = interfacial (surface) tension
d = diameter of pipe	p = pressure

Although these groups describe the mechanisms affecting two-phase flow, their application is not routine. Since the phases move at different velocities, what value of "v" does one use in a given group? Also, in a given situation what is the relative magnitude of the forces represented? Does one group control and therefore become suitable for correlation, or must several be used?

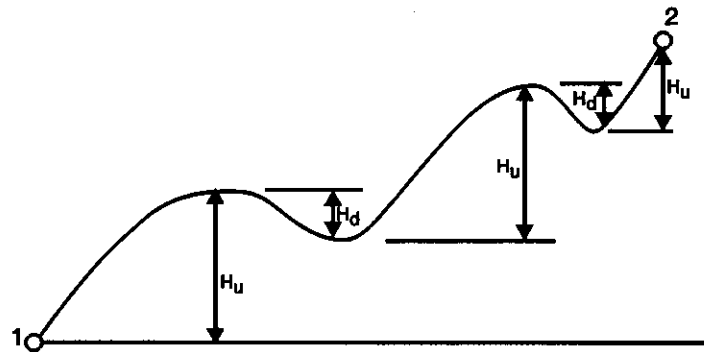
Different investigators have answered these questions in a different manner. In a two-phase system some composite or effective values are needed since each phase possesses different values for each term. Therefore, it must be remembered that all correlations are based on the same fundamental principles; they differ only in how questions like those above are answered in fitting available data.

In spite of the correlation accuracy that might be reported for a given correlation, we recommend assuming an accuracy of 15-20% in design of a new system. In planning, this leads to decisions which are economically sound and provide necessary operating flexibility. How good are the flow rate, pressure and property data to be used with a correlation in design? They are probably no more accurate than the correlation. So ... as a practical matter a proper correlation is as good as the data being inserted into it.

HORIZONTAL TWO-PHASE FLOW

No line buried in the ground or under water is truly horizontal. The word "horizontal" simply means that the length is a much larger number than any elevation change. In such lines two-phase flow occurs uphill and downhill as well as horizontally. Some aspects of vertical flow are occurring simultaneously with said horizontal flow.

The following figure is a view of a typical line where the magnitude of the uphill and downhill portions has been obtained by survey. Unlike single-phase flow, each change in elevation between (1) and (2) affects head considerations.



If the energy of the liquid entering a low spot is insufficient to carry it over the next "hill," liquid will collect in the low spot as shown next.



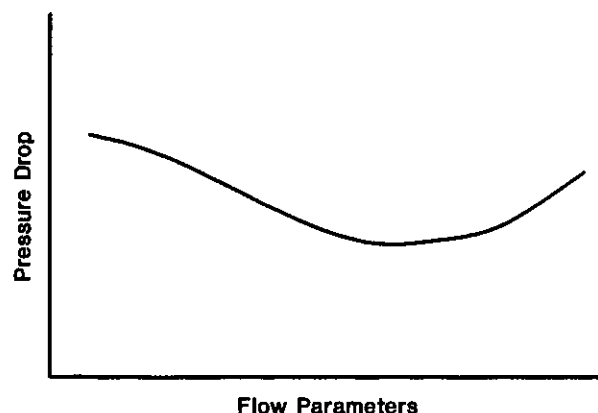
As this liquid builds up, the opening for gas decreases. At some point there is sufficient pressure force to start the liquid moving. A portion is gas lifted over the hill and the remainder slips back. The liquid flowing on contributes to liquid buildup in the next low spot. Then, in time, it surges. This process is repeated successively as liquid moves down the line.

This represents an inefficient use of pressure energy. It takes energy to overcome liquid inertia and produce the *momentum* needed. Furthermore, the lifting process is relatively inefficient. Both contribute to excess pressure loss. For this reason, it is desirable to maintain high enough flow rates to keep the liquid in motion.

In addition to excess pressure loss, liquid collection causes excess liquid holdup and results in larger liquid surges at the outlet of the line.

Both excess pressure drop and surges are noted in lines operating well below design capacity. As rate increases, both decrease. So ... the proper motto is, do not oversize your lines. Think small!

This is illustrated by the general figure below. Pressure loss goes through a minimum and then increases as flow rate increases. The reasons include the liquid segregation discussed above.



For the typical offshore installation where initial flow rates are low, "slug catchers" and other equipment must be designed for this worst case. The line must, of course, be sized for the maximum anticipated rate, but some consideration must be given for lower rates so that the system is satisfactory under all conditions expected.

Basic Correlations

All correlations are some form of a *momentum balance* and *continuity equation* containing terms for the head, friction and acceleration effects. These vary with the flow regime. Thus, it is common to write a different equation for each specific regime.

$$\left(\frac{dP}{dL} \right) = \underbrace{\frac{f_{tp} \rho_{tp} v_{tp}^2}{2 g_c d}}_{\text{friction}} + \underbrace{\frac{g \rho_{tp} \sin \Phi}{g_c}}_{\text{elevation}} + \underbrace{\left(\frac{\rho_{tp} v_{tp}}{g_c} \right) \left(\frac{d v_{tp}}{d L} \right)}_{\text{acceleration}} \quad (10.58)$$

Equation 10.58 includes several assumptions: 1) steady state, 2) no nuclear reactions, 3) no pumps or compressors in the system, and 4) adiabatic. With these limitations, this provides the basis of most fluid flow calculations.

The complexity of the solutions has increased with the development of computer models. The various correlations now available can be placed in three general classes.

Class 1

- a. Liquid hold-up is not considered in density.
- b. Liquid hold-up and wall friction losses are incorporated into a friction factor.
- c. No distinction is made between flow regimes.

Class 2

- a. Liquid hold-up is considered in density.
- b. Liquid hold-up may be correlated separately.
- c. Friction factors are based on composite properties.
- d. No distinction is made between flow regimes.

Class 3

- a. Density is adjusted for liquid hold-up.
- b. Liquid hold-up is estimated using some concept of slip velocity (difference between gas and liquid flow rates).
- c. The continuous fluid phase(s) determine(s) wall friction losses.
- d. Flow regimes are considered.

Most of the correlations in the past twenty years are in Class 3 and are only applicable as computer solutions.

As noted in Reference 10.1, there is a group of correlations commonly used for horizontal flow. The usual, intelligent approach is to make the calculation by several methods to establish a likely performance range. No one method is superior to all others for general usage.

One of these methods (Flanigan's) is suitable for manual calculations and illustrates the relative role of some key variables.

Modified Flanigan Correlation^(10.25)

This work was done on field systems and has proven useful even though it is relatively simple. The relationship between gas flow rate, diameter and pressure drop is represented by the Panhandle A equation (Table 10.1). Two corrections are made for two-phase flow.

1. The value of outlet pressure (P_2) is adjusted for the pressure loss due to uphill and downhill flow of two phases, including the effect of *holdups*.
2. The efficiency term (E) is correlated to reflect measured system performance based on gas velocity and liquid-gas ratio.

As thus modified the Panhandle A equation becomes

$$q_{sc} = K (T_{sc}/P_{sc})^{1.0788} \left[\frac{P_1^2 - (P_2 + \Delta P_2)^2}{T_m L z_m} \right]^{0.5394} (1/\gamma)^{0.4606} (d)^{2.6182} E_{tp} \quad (10.59)$$

Where:

q_{sc} = gas rate at T_{sc} , P_{sc}
 P = absolute pressure
 P_{sc} = pressure, standard conditions
 T_m = mean absolute temperature of line
 T_{sc} = temperature, standard conditions
 d = inside diameter of pipe
 L = pipe length
 γ = gas relative density
 z_m = mean compressibility factor
 E_{tp} = two-phase efficiency
 K =

Metric	English
m^3/d	scf/d
kPa	psia
kPa	psia
K	°R
K	°R
m	in.
m	mile
—	—
—	—
1.198×10^7	435.87

The term $(P_2 + \Delta P_2)$ is the outlet pressure of a truly horizontal line equivalent to that of the actual P_2 .

The pressure correction for inclined flow is calculated from Equation 10.60

$$\Delta P_2 = A (\rho_L E_h \Sigma H_u - \rho_g \Sigma H_d) \quad (10.60)$$

Where:

ΔP_2 = additive correction to Panhandle A P_2
 A = constant
 ρ_L = liquid density
 ρ_g = gas density
 H = head
 E_h = empirical head factor (Figure 10.8)
 H_u = uphill heads
 H_d = downhill heads

Metric	English
kPa	psi
0.009 81	0.0069
kg/m^3	lb/ft^3
kg/m^3	lb/ft^3
m	ft
—	—
m	ft
m	ft

The head terms are found from the line profile as shown previously.

Figure 10.8 is a correlation to find the liquid head factor in Equation 10.60. *Superficial gas velocity* is the actual volumetric rate divided by cross-sectional area; it is assumed that gas occupies all of the pipe. At low velocity the head loss is high. Liquid collects in the low spots and is lifted periodically. This is inefficient. As velocity increases above 2 m/s [6.6 ft/sec] this head loss decreases rapidly. Above about 8 m/s [26.2 ft/sec] there is little effect of velocity on head pressure loss.

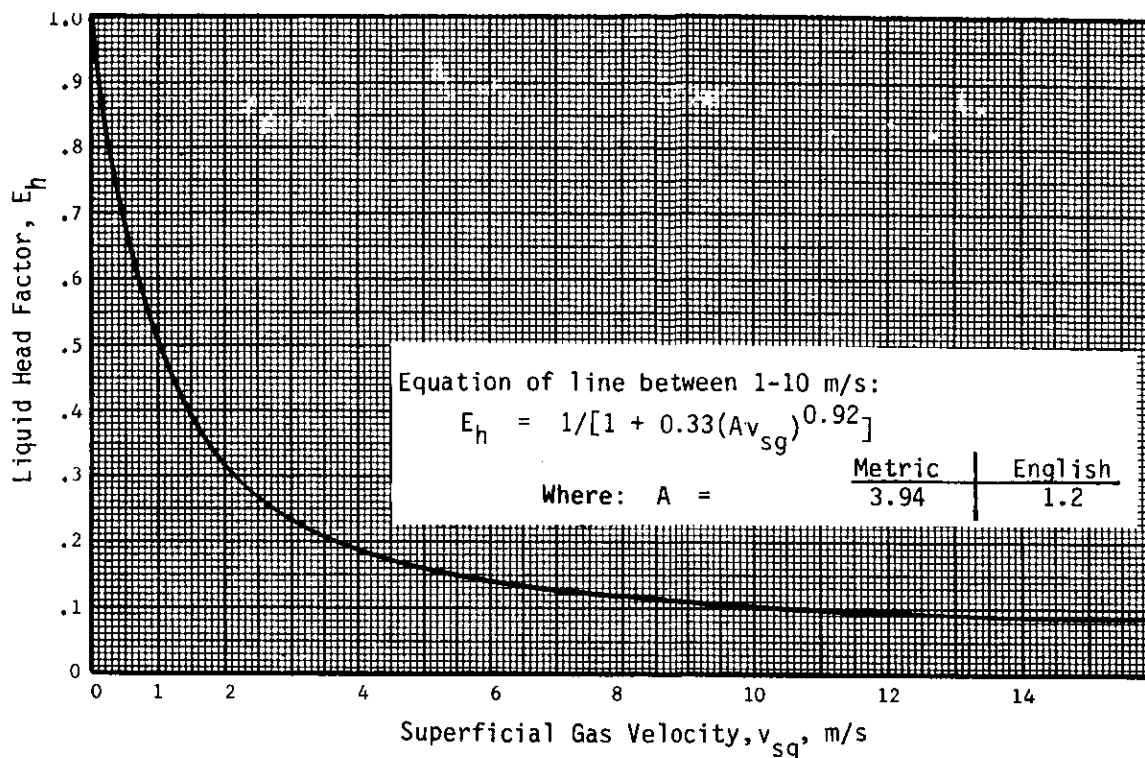


Figure 10.8 Flanigan's Liquid Head Factor Results.^(10.25)

The velocity for use in Figure 10.8 may be calculated from Equation 10.61.

$$v_{sg} = (A) \frac{q_{sc} z_m T_m}{d^2 P_m} \quad (10.61)$$

Where:

q = gas flow rate (sc)
d = internal pipe diameter
P_m = mean pressure in section
T_m = mean temperature in section
A = constant for conversion
v_{sg} = gas velocity
z_m = compressibility at T_m and P_m

Metric	English
10 ⁶ m ³ /d	10 ⁶ ft ³ /d
m	in.
kPa	psi
K	°R
5.12	60.0
m/s	ft/sec

The two-phase "efficiency" (E_{tp}) is found from Figure 10.9. The v_{sg} on the abscissa is the same one used for Figure 10.8. The "R" is the liquid-gas ratio in the pipeline segment.

At the upper right end of this curve the line contains a relatively small amount of liquid. The efficiency approaches that of a gas line. To the left, the amount of liquid increases and flow behavior approaches bubble and plug flow.

In many lines, the liquid/gas ratio, R, varies throughout the length. As the gas cools, R increases. This cooling follows normal temperature reduction patterns. On a long line, the value of R quickly becomes stable. If one only knows R at the inlet and outlet of such a line, using R at the outlet is the best of the two values since it probably exists for most of the line length.

Because of the variation of R, P and T with length, a long line is sometimes divided into segments for calculation purposes. An average or mean P and T is found for each section. R is found from an equilibrium

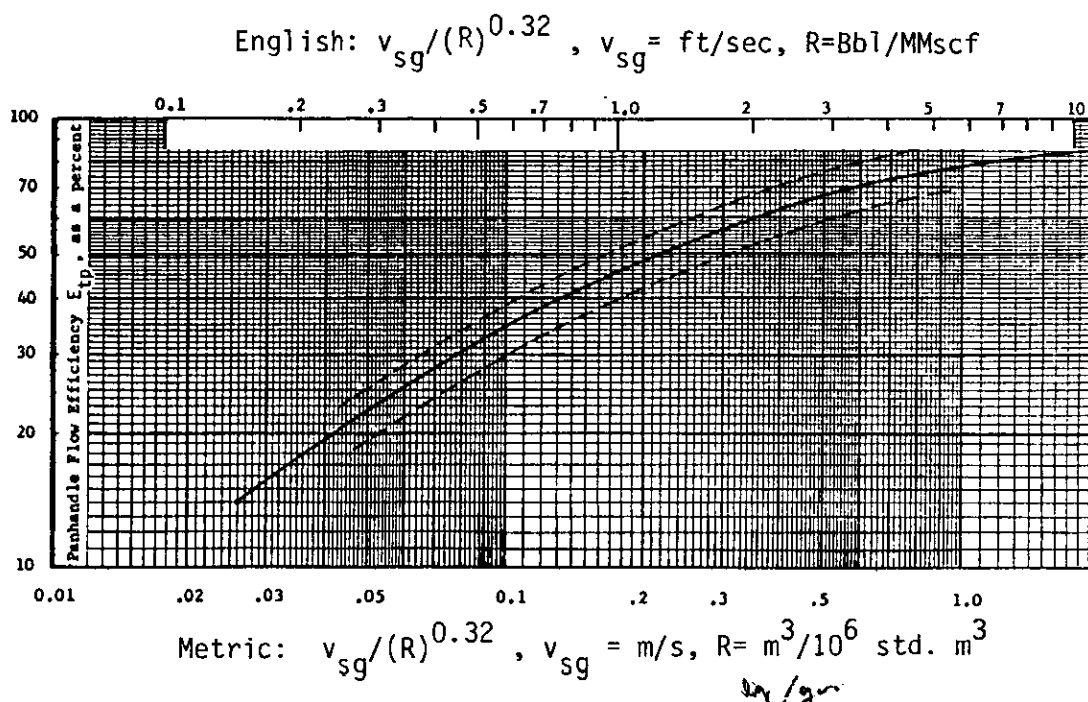


Figure 10.9 Flanigan's Holdup Correlation

calculation at these values. The ΔP of the segment is found after determining its regime and appropriate equation. If numerous segments in series are used, the calculation quickly becomes too time-consuming to be done manually.

Three different basic calculations are made using this method.

1. *Flow rate and diameter known, solve for pressure drop.* – This is a straight-forward solution. Solve Equation 10.59 for $(P_2 + \Delta P_2)$. P_2 is the desired outlet pressure – or if P_2 is fixed, solve equation for inlet pressure P_1 .
2. *Flow rate and pressure drop known, solve for diameter.* – Assume a diameter to start a trial-and-error process. For this assumed value calculate ΔP_2 and E_{tp} and solve Equation 10.59 for "d." When value calculated checks the assumed value (within even pipe sizes), the correct diameter is known.

What is a good first value of "d" to assume? Use Equation 10.61 for an assumed superficial velocity of 8 m/s [26 ft/sec] to solve for "d." This same assumption is used also in Figures 10.8 and 10.9 for the first trial.

3. *Pressure drop and diameter known, solve for flow rate.* – This also is trial-and-error. Assume a flow rate. This fixes a velocity for the first trial. When the assumed value agrees with the value calculated in Equation 10.59 with about 10%, you have a solution.

One can guess a velocity as in (2) above and use Equation 10.61 to find a corresponding "q" as a first guess.

This is a simple trial-and-error process because many of the equation values stay constant for all trials.

This modified Flanigan approach has proven useful for lines where the efficiency calculated is above 50%. Below this the combination of gas velocity and liquid loading is such that a more complex system tends to exist. This correlation certainly is valid only when one has the expectation that gas is present as a continuous system. Toward the lower end of the curve this is less likely.

Figure 10.10 shows results reported by Gould and Ramsey^(10.26) for a 15-inch line containing a 0.7 relative density gas and a 0.83 relative density oil at various liquid-gas loadings. The pressure used was 6.9 MPa [1000 psig] and the temperature was 4°C [40°F]. The results should not be regarded as typical. They are shown primarily to illustrate the spread of results one can obtain by various methods. Reference 10.1 shows other test results.

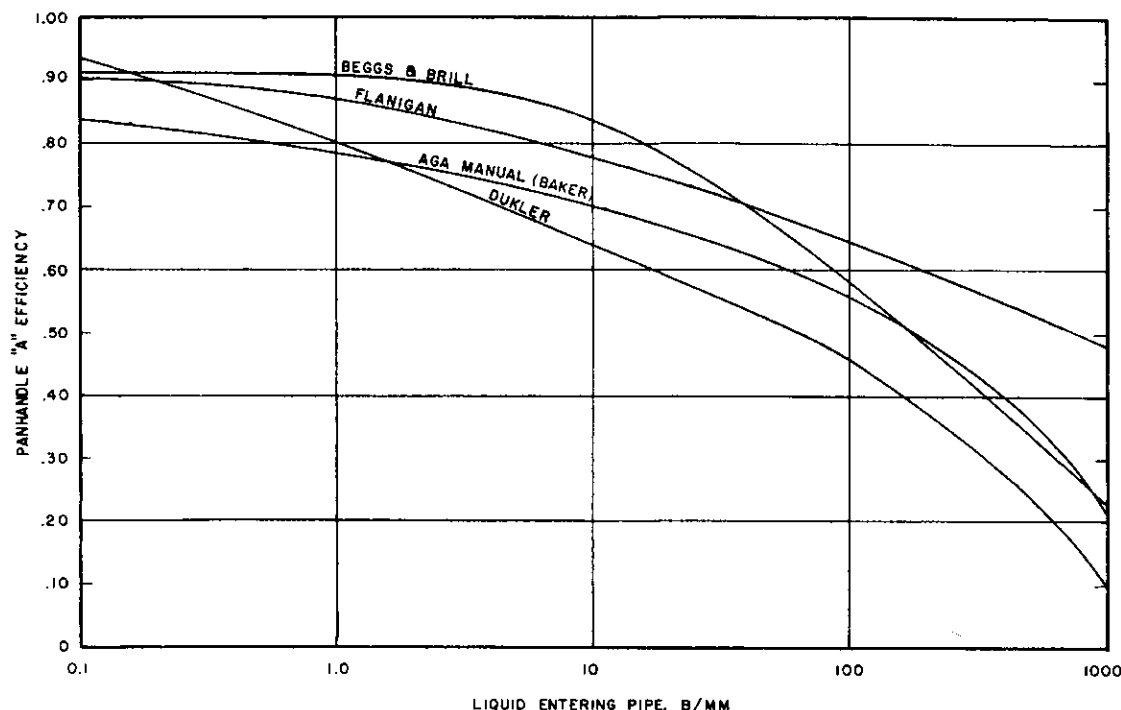


Figure 10.10 A Comparison of Calculation Methods for One Case

VERTICAL TWO-PHASE FLOW

The principles are the same as for horizontal or inclined flow except that gravitational forces are acting in an opposite direction to flow. The segregation patterns are different. Figure 10.11 illustrates the effect of the variables on vertical flow behavior.

The values of RN and N are two of the four dimensionless groups used to characterize system behavior.

$$RN = \text{gas velocity number} = v_{sg} (\rho_L / g\sigma)^{0.25}$$

$$N = \text{liquid velocity number} = v_{sL} (\rho_L / g\sigma)^{0.25}$$

Where:

- v_{sg} = superficial gas velocity
- v_{sL} = superficial liquid velocity
- ρ_L = liquid density
- σ = surface tension
- g = gravity acceleration

Since RN and N are dimensionless numbers, any consistent set of units may be used so long as the result is dimensionless. The gas velocities shown are found by dividing the volumetric flow rate of each phase by the total pipe area.

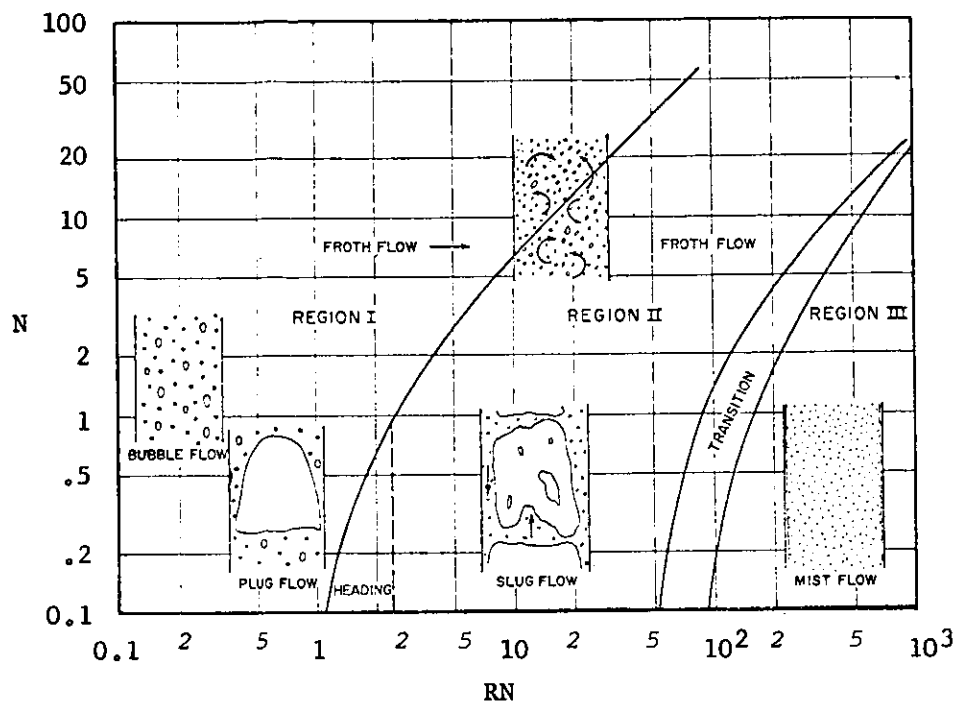


Figure 10.11 One Correlation of Vertical Flow Regimes

In Region I, at low gas numbers and high liquid numbers, one encounters a liquid with gas bubbles in it, so long as the gas-oil ratio is relatively low and the flowing pressure gradient primarily is the static head plus liquid friction loss. The relative magnitude of friction loss depends on flow rate.

For liquid rates less than 0.4 m/s [1.3 ft/sec], increased gas flow causes the bubbles to combine and form plugs. As gas flow increases further these plugs collapse and form slugs. In these regions wall friction is rather negligible.

If v_{sL} is still less than 0.4 m/s [1.3 ft/sec] but v_{sg} is about 15 m/s [49 ft/sec], or greater, the slug flow of Region II changes to mist flow of Region III. At this point the gas becomes the continuous phase with the liquid in droplet form and as a film along the wall. In Region III wall friction is a major factor in pressure loss. The liquid on the surface occurs in "ripples" which affect wall friction. The magnitude of these ripples (and thus wall friction) increases dramatically with gas flow rate. Liquid holdup in Region III is small.

Froth flow which occurs across the lines of Regions I and II occurs at high liquid velocities. Duns and Ros expect it to occur when v_{sL} is greater than 1.6 m/s [5.2 ft/sec]. At such rates no plug or slug flow was observed. No set flow pattern can be discerned. Only at high gas flows does separation of phases become discernible.

Regardless of the specific correlations used for calculation of a particular vertical system, Figure 10.11 and the implications of it provide a real insight into what is going on in the vertical pipe.

What are the primary factors? Liquid and gas velocities, pipe diameter, fluid viscosity, liquid surface tension and fluid density. Gas-oil ratio is a factor but may be represented (indirectly) by the superficial velocities and diameter. Stated simply, the type of flow is governed by fluid physical properties, their relative quantities and the degree of turbulence and other forces affecting fluid distribution in the pipe.

A group of correlations equivalent to those for horizontal flow are available for vertical flow. One of the earlier ones (Poettmann and Carpenter) is suitable for manual calculation.^(10.27) This correlation works well for crude oil streams with a gas-oil ratio to about 180 m³ gas/m³ liquid [1000 scf/bbl] when the oil viscosity is less than 10 cp.

The friction factor plot shown in Figure 10.12 was prepared with the ordinate "f" as the composite friction factor; the abscissa as the numerator of Reynolds number, $dv\rho$.

$$dv\rho = \frac{A q_o m_o}{d} \quad (10.62)$$

Where:

q_o = oil production
 m_o = $\frac{\text{total mass of fluids}}{\text{volume of oil}}$
 d = pipe diameter
 A = unit conversion constant

Metric	English
m^3/d	bbl/day
kg/m^3	lbm/bbl
m	ft
9.88(E-06)	1.47(E-05)

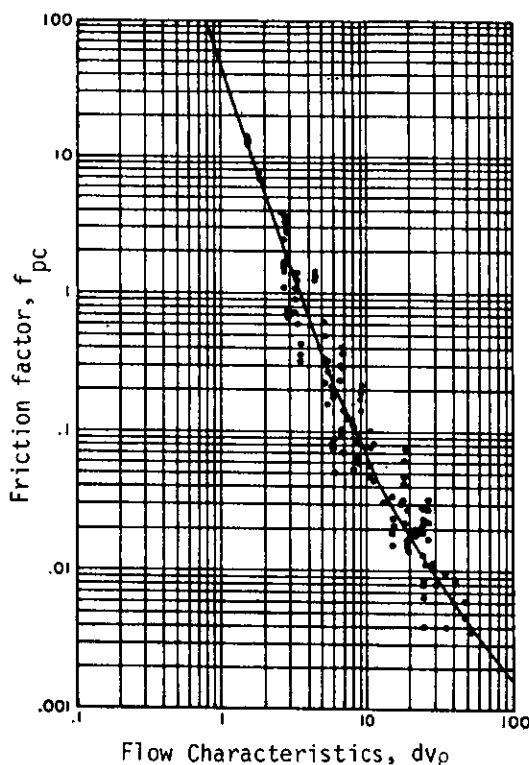


Figure 10.12 Poettman-Carpenter Friction Factor Correlation

The value of " m_o " may be found as:

$$m_o = A \left[\gamma_o + \left(\frac{q_w}{q_o} \right) (\gamma_w) \right] + (GOR) \rho_a \gamma_g \quad (10.63)$$

Where:

γ_o = relative density of oil
 γ_w = relative density of water
 γ_g = relative density of gas
 q_o = oil production
 q_w = water production
 GOR = gas/oil ratio
 A = conversion constant
 ρ_a = density of air at standard condition

Metric	English
—	—
—	—
—	—
m^3/d	bbl/day
m^3/d	bbl/scf
std m^3/m^3	scf/bbl
1000 kg/m^3	350 lb/bbl
1.21 kg/m^3	0.0764 lb/ft ³

With Equations 10.62 and 10.63, and Figure 10.12, one then can determine the Poettmann-Carpenter friction factor (f_{pc}). It is related to pressure loss by the equation

$$\Delta P / \Delta L = \rho_m (g/g_c) + \frac{f_{pc} q_o^2 m_o^2}{A \rho_m d^5} \quad (10.64)$$

Where: ΔP = total pressure loss
 ΔL = length of pipe
 ρ_m = mixture density
 d = pipe I.D.
 g/g_c = force equivalent
 A = unit conversion constant

Metric	English
Pa	lbf/ft ²
m	ft
kg/m ³	lbm/ft ³
m	ft
9.81	1.0
2.09(E+09)	7.41(E+10)

Although temperature does not occur directly in the above procedure, some average value must be known (or assumed) to calculate some of the variables.

Equation 10.64 is in oilfield terms but it is derived directly from the energy balance for fluid flow. The first term is the elevation term and the second term is the friction loss term. The equation could be written as

$$\Delta P = (g/g_c)(\rho_m)(\Delta L) + \frac{2 f_{pc} (\Delta L) v_m^2 \rho_m}{g_c d} \quad (10.65)$$

The only difference between this equation and the single phase version is the use of mean density and velocity (along with a special friction factor plot). Mean density is the mass of total fluids flowing divided by the total volume of fluids flowing at a mean P and T. The constant in Equation 10.64 merely is the summation of all numbers to convert from velocity to a mass term and the numerical constants.

This calculation has been reduced to a series of charts. Figures 10.13 and 10.14 are for 2-inch and 2.5-inch tubing, respectively. The dashed lines are for a flowing density of 80 kg/m³ [5 lb/ft³], which show a reversal of position from the other data.

The use of this procedure is illustrated by Example 10.14 taken from Reference 10.28.

The same result could have been obtained from Figure 10.14. Multiply 302 times 343 to obtain 103 600 pounds of total fluid per day. Enter the left ordinate at about 103, proceed horizontally to the curved line for an average density of 40 lb/ft³. Read vertically to lower abscissa to read about 0.285 psi/ft, checking the calculation.

Like all such calculations, one needs an average pressure and a temperature for the line segment being calculated to determine fluid properties. A useful correction for estimating temperature is Figure 10.15. The geothermal gradient in the area usually is known.

In using this correlation, calculate the total oil and water flow rate in m³/day [bbl/day]. Proceed vertically to the appropriate geothermal gradient and read horizontally to the ordinate to find the *flowing temperature gradient* in °C/100 m [°F per 100 feet]. This number can be used to estimate average temperature for a given segment or over the entire wellbore.

Average pressure can be found by trial-and-error by assuming a value, calculating total ΔP , and then checking to see if the calculated mean pressure assumed checks the one calculated. If not, iterate until it does.

Sometimes in the early stages one does not know very much about the oil being handled. Figure 10.16 from Standing may prove useful for estimation purposes.

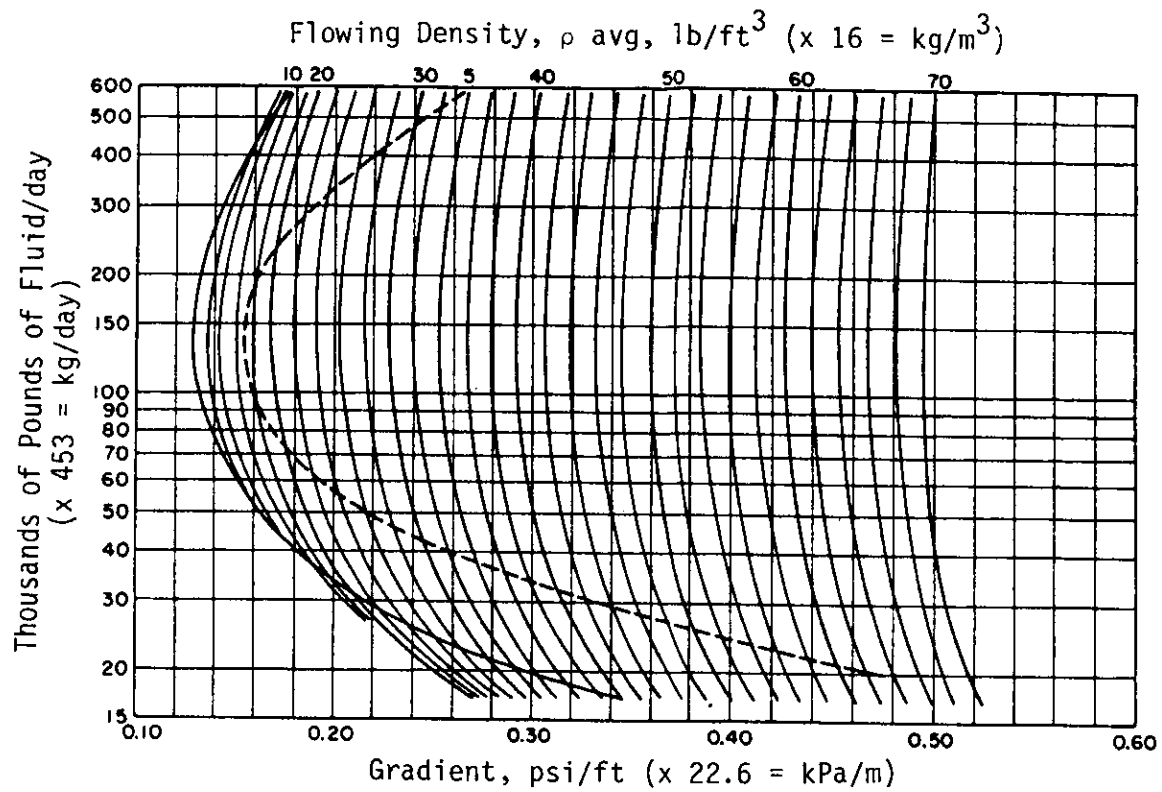


Figure 10.13 Pressure Gradients for 0.0507 m [1.995 in.] I.D. Tubing

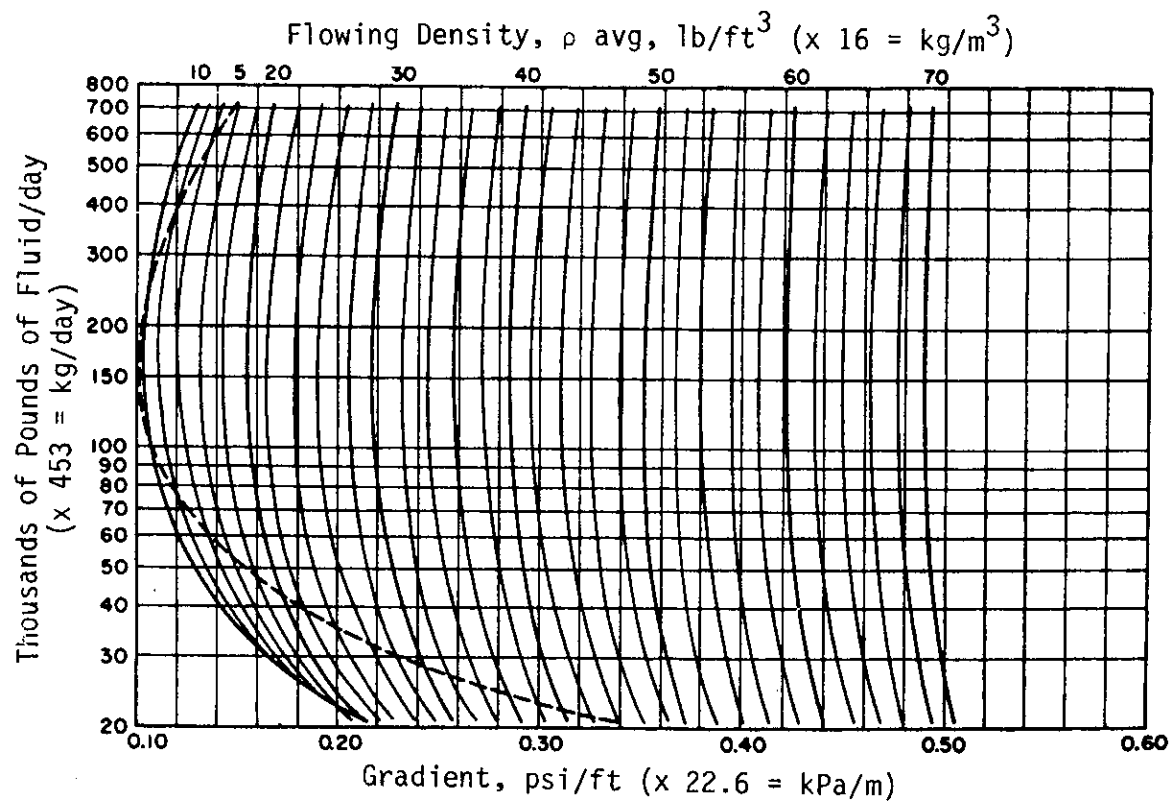


Figure 10.14 Pressure Gradients for 0.0620 m [2.441 in.] I.D. Tubing

Example 10.13: Correlate the field data of a flowing well and calculate pressure gradient, given:

Oil production = 48 m³/d [302 STB/day]
 Relative density of st oil = 42.3°API (st refers to "stock tank")
 Gas-oil ratio = 167 m³/m³ [936 scf/stb]
 Relative density of gas = 0.816
 Tubing size = 0.062 m [2.441 in. I.D.]
 Tubing pressure = 5.9 MPa [860 psia]
 Flowing bottom-hole pressure = 17.4 MPa [2530 psia]
 Depth to middle of perforations = 2088 m [6850 ft]
 Average flowing temperature = 38°C [100°F]
 No water production
 Average density = 640 kg/m³ [40 lbm/ft³]

English: The relative density of the stock tank oil at 60°F is

$$\gamma_o = \frac{141.5}{131.5 + 42.3} = 0.814$$

Substituting in Equation 10.63, the weight rate of flow is

$$m_o = (350)(0.814) + 0.0764(0.816)936 = 343 \text{ lbm/st Bbl}$$

From Equation 10.62:

$$d v \rho = \frac{(1.47 \times 10^{-5})(302)(343)}{2.441/12} = 7.51$$

From Figure 10.12:

$$f_{pc} = 0.13 \text{ (approximately)}$$

From Equation 10.64:

$$\begin{aligned} \Delta P_{AL} &= 40 + \frac{(0.13)(302)^2(343)^2}{(7.41 \times 10^{10})(40)(2.441/12)^5} \\ &= 41.4 \text{ lbf/ft}^2/\text{ft} = 41.4/144 = \underline{0.29 \text{ psi/ft}} \end{aligned}$$

Metric:

$$d v \rho = \frac{(9.88 \times 10^{-6})(48)(982)}{0.062} = 7.51, \quad f = 0.13$$

$$m = (1000)(0.814) + (168)(1.21)(0.816) = 982 \text{ kg/m}^3 \text{ oil}$$

$$\Delta P_{AL} = 640(9.81) + \frac{(0.13)(48)^2(982)^2}{(2.09 \times 10^9)(640)(9.16 \times 10^{-7})} = 6.514 \text{ kPa/m}$$

The Poettmann-Carpenter method is an example of flow where liquid is definitely a continuous phase and gas probably is discontinuous. This is really a form of *distributed* flow rather than intermittent flow.

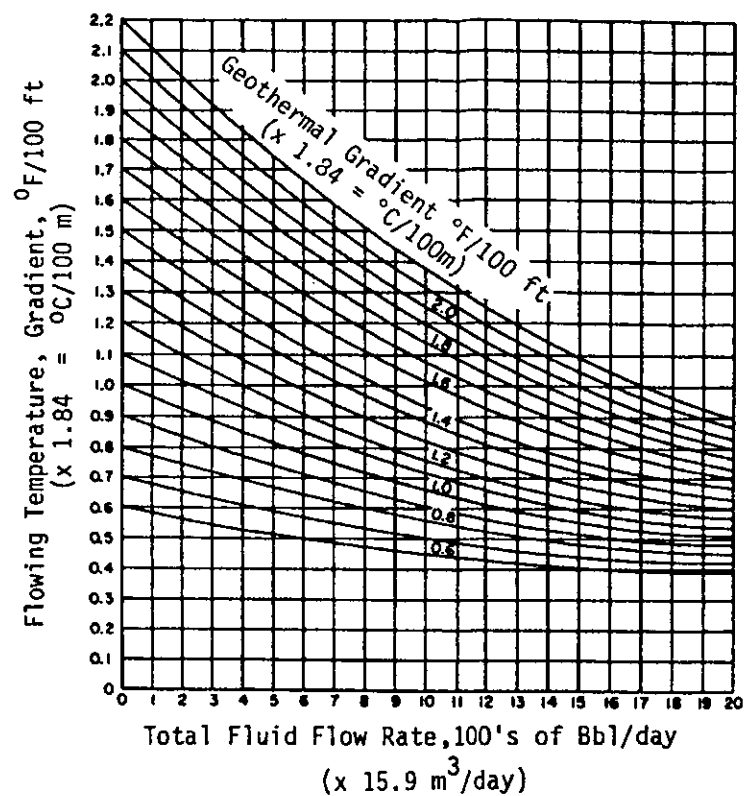


Figure 10.15 Flowing Temperature Correlation.

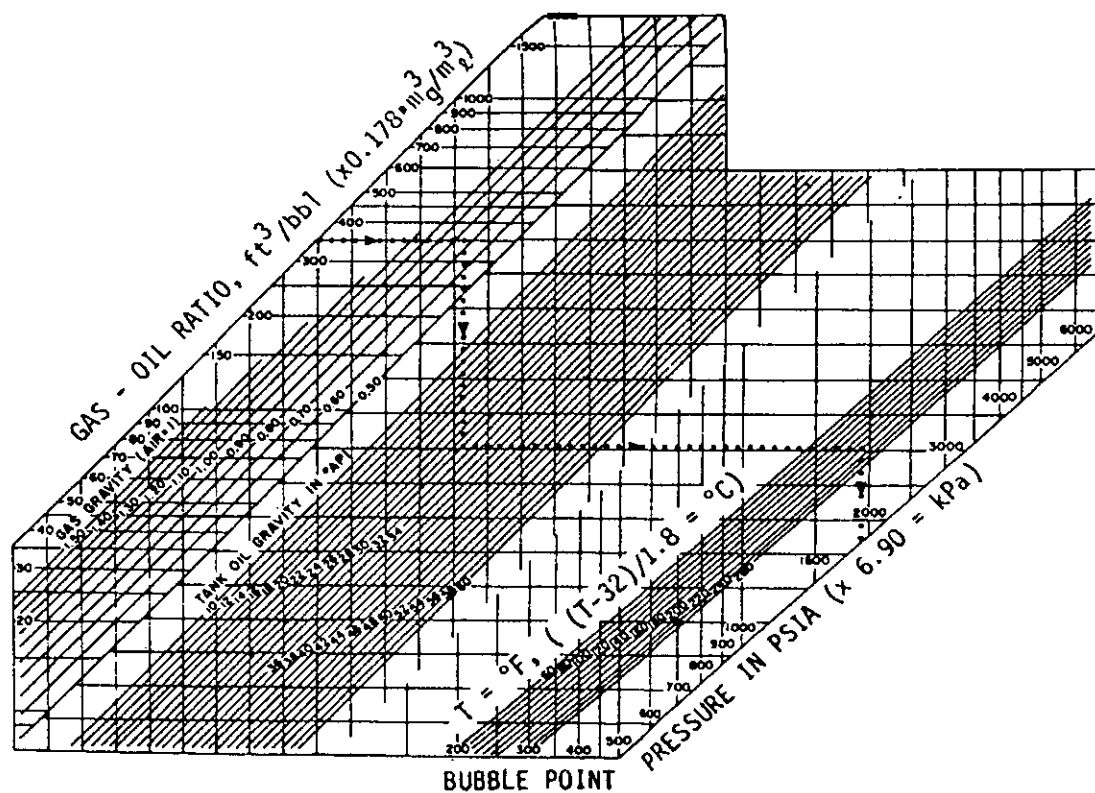


Figure 10.16 Standing's Solution Gas-Oil Ratio Correlation

THE CALCULATION STRATEGY

The two methods outlined herein are only two among many correlations available. The situation is much like PVT equations of state. All are empirical and use the same fundamental principles. They differ only in the manner in which they handle variables and the data used to obtain their numerical value. Names like Beggs and Brill, Martinelli, Dukler, Guzhov, Orkiszewski, etc., are used on different calculation models. These serve as an equation format which may be modified to fit data on a specific system.

No one model is inherently superior. Insofar as I am concerned, the strategy that should be followed is expressed by the opinions I added to Reference 10.1 in the process of preparing it for publication. A version of that follows.

Horizontal Flow

The relative amounts of the two phases and their velocities are of fundamental importance. The basic method of calculation depends on these, for they determine to a large degree the manner in which the phases interact (flow regimes).

Although a number of different flow regimes have been identified (directly or indirectly), the number probably exceeds true calculational ability. Having a large number of regimes enhances the researcher's chances of obtaining an empirical data fit (by working backward). But then, working the results forward (for real systems) poses problems. This is indicated indirectly by the fact that different investigators cannot agree on where to place the lines between different regimes.

It seems most logical to contend with only three basic flow regimes:

Distributed (mist, bubble, froth). – Where there is only one continuous phase, the other phase(s) being distributed throughout this phase as drops or bubbles.

Segregated (stratified, wave, annular). – Both phases present are continuous with the gas phase moving faster with a resultant shear force at the interface.

Intermittent (slug, plug). – Continuity of both phases is interrupted and a series of inertial and kinetic force changes occur throughout the line section.

The latter is, of course, most difficult because the overall performance of the line segment is the summation of a series of intermittent, somewhat random occurrences at various points within that segment.

Transition zones do occur but by the very definition of the word "transition" these are temporary, unstable regions that should not last very long for a properly operated line. For this reason, they should have little design application.

As a design factor, holdup is a very general term that tries, but cannot describe localized behavior within a segment. Yet, it is about all one currently can do. If it is assumed that over a finite period liquid plus gas out equals that coming in, the value of this number can be estimated from specified or calculated values. At least tentatively one can establish the likelihood of one of the three regimes above.

If the relative quantity of the continuous phase is above 80% and that phase is in *fully turbulent flow*, it is difficult to envision anything other than distributed flow as the predominant regime. Full turbulence would be determined by the usual criteria for gas and liquids, as the continuous phase.

What about liquid holdups from 20-80%? Intermittent flow is difficult to achieve unless the gas is in *fully turbulent flow*. The magnitude of the shear forces and the axial forces present will tend to produce some degree of intermittent flow as the "waves" formed begin to have an amplitude equal to pipe diameter.

It is probable that true segregated flow occurs only when the gas velocity (based on the fraction of the total area available to it) is in *partially turbulent flow* only. The degree of this most certainly depends on pipe diameter and fluid properties.

If one does not have distributed or segregated flow conditions available, the flow is intermittent by default. This is obvious, but the implications may not be. These are discussed in the next section.

Fluid Properties. – It is routine to calculate density, viscosity and surface tension for a given phase. But, energy balances require composite properties for separate phases. The best one can do is calculate some kind of mean number that fits the correlation being used. This probably is not critical concern since the P and T used to obtain the value may contain more error than the calculation mode itself.

Line Size. – This obviously is a factor. Pipe smoothness increases and friction drop becomes relatively less as pipe diameter increases. As the diameter gets larger the liquid slugs also get larger, by the very geometry involved. The waves in segregated flow must get higher to occupy the full cross-section.

In spite of this, it is interesting that data from small diameter pipes can be effectively applied to large diameters if the model used properly identifies the relative magnitude of the various forces involved. Diameter effects are represented by the frictional, kinetic, acceleration and gravity forces in the model.

Inclination. – Uphill sections can exhibit all flow regimes. In the downhill section segregated flow normally will predominate if the volume of this section is comparable to the preceding uphill section. Gravity will separate the phases. If this is true, the pressure recovery probably can be represented reasonably by the gas static head involved.

Pressure. – This effect shows up in both flow rate and fluid property calculations. However, scaling low pressure data to high pressures introduces a judgmental problem. For example, the density of gas changes far more rapidly with pressure than liquid. At 6.9 MPa [1000 psi] the typical natural gas is about 80-85% denser than at atmospheric pressure. Liquid density does not change very much. Thus, gas shear and impact forces change drastically, among other things.

Once again, though, the application of sound models based on lower pressure data to higher pressure systems does not appear to introduce serious error.

Temperature. – Like pressure, this has an obvious effect. Calculation of a temperature profile is an approximation at best. Lines are coated and may be covered with concrete; water currents and surrounding soil characteristics vary. The method of temperature prediction is from actual data on similar systems (preferably under the same basic surrounding conditions).

PVT Behavior. – A prediction is necessary for the real system. In almost all research studies, deliberately picked stable liquids eliminated this problem.

Of concern is changing liquid/gas ratios and fluid properties. Line segments must be chosen so that the change in these values is not so dramatic in that section as to compromise the calculations being made. The line segments need not be of equal length. Once the temperature approaches that of the surroundings, PVT behavior is relatively insensitive to subsequent changes in pressure, in most cases.

Vertical Flow

The same factors apply, although the character of flow is somewhat different because gravity is working opposite to flow forces instead of at right angles to them. There really are only two basic flow regimes – distributed and intermittent. One can obtain a liquid ring around the pipe but this is either a transition zone or a rather thin liquid ring with mist flow.

Distributed flow offers no particular calculation problem. Several methods are available which give useful results. Slug flow is a problem, both from a design viewpoint and operations. No one method is completely adequate for slug flow design.

APPLICATION OF THE TECHNOLOGY

It is no trick to design a two-phase line for a single flow rate with fixed gas/oil ratios and fluid properties. It could be sized so that flow is rather steady and liquid surging is handled easily with rather ordinary equipment. If flow is distributed, the design accuracy may be 5 to 10%. At worst, for any flow regime, it may be 15 to 20%. All of these are adequate for economic design.

The problem is created by the usual real situation that occurs. Early on in most cases, the line will be operating at less than its final design rate. Often it is not even certain at this point what the final rate will be. Also, reservoir characteristics dictate that the gas/oil ratio will change with time. So ... in the real world this is an exercise in uncertainty. The input quantities may be far more uncertain than the calculations employed. What does one do to reduce the risk to an acceptable level?

The first step is to produce design specifications that are a realistic picture of the future. These cannot be obtained in a committee meeting or independently by a contractor. They must involve input by geological, reservoir, production and even marketing specialists. Needed are forecasts of development patterns, performance expectations and the like. Required is a strategy for line development. Offshore, the line to shore will be installed early so it must reflect all of these factors. Early in its life it will be inefficient and possibly exhibit higher than normal instability. Predicting pressure loss is only a part of the problem. Equipment at one or both ends of the line must be able to handle the fluctuating pressures and liquid expected at all stages in line life. *Pigging* may be required during early life to compromise the problem.

There are numerous programs, correlations, and "experts" around to "help" the engineer calculate what is one of the more difficult yet common problems – two-phase flow calculations. Most experts will spend a great deal of time explaining why their favorite method is actually giving the correct answer even though it doesn't match the real world.

If this early planning is done intelligently and professionally, the stage is set for the next step – the engineering calculations. There are three basic strategies one can follow.

1. Pick your (or the company's) single favorite calculation method and presume it will be right.
2. Choose several of the "popular" calculation methods and use them to develop a range of results.
3. Compare your system's characteristics with various documented data and choose several methods that have been tested on comparable systems.

These are listed in order of usual preference. The first is closer to religion than engineering. There is no one best method for all applications. If (1) is used, more luck than skill is involved.

Table 10.7 gives some notes on some of the commonly available systems as a general guide. This table should not be taken as all inclusive or an endorsement; it is more a list of those systems "commercially available."

Both (2) and (3) may be satisfactory but (3) is the best approach. The system characteristics may indicate use of a given correlation in the suite of calculations, even though it is not widely used. The suite probably should use three calculation approaches, throughout the range of flow conditions expected for different diameters of lines. As in single-phase lines, more than one diameter of line will work. Unlike single-phase lines, though, over-sizing should be avoided. It is critical to have a high enough gas velocity to keep the liquid moving continuously. So ... think small. This tends to compromise the changing flow rates during line life.

TABLE 10.7
Two-Phase Pressure Drop Correlations

Correlation	Notes
Vertical Upward Flow	
Duns & Ros	Good in mist and bubble flow regions.
Angel-Welchon-Ross	Applicable for high flow areas and annulus flow. Recommended for high volume wells and low gas/oil ratios.
Hagedorn & Brown	Best available pressure drop correlation for vertical upward flow. Most accurate for angles of inclination greater than 70 degrees.
Orkiszewski	Results reliable for high gas/oil ratios. Most accurate for angles of inclination greater than 70 degrees.
Aziz	Generally slightly overpredicts pressure drop; other correlations tend to underpredict. This fact can be used to bracket the solution. Most accurate for angles of inclination greater than 70 degrees.
Beggs & Brill	Good for all angles of inclination. Predicts the most consistent results for wide ranges of conditions.
Gray	Specifically designed for condensate wells (high gas/oil ratios). Recommended ranges: velocity < 15 m/s [50 ft/sec]
Horizontal Flow	
Lockhart-Martinelli	Widely used in the chemical industry. Applicable for annular and annular mist flow regimes if flow pattern is known a priori. Do not use for large pipes. Generally overpredicts pressure drop.
Eaton	Do not use for diameters < 50 mm [2 in.]. Do not use for very high or very low liquid holdup. Underpredicts holdup for $H_L < 0.1$. Works well for $0.1 < H_L < 0.35$.
Dukler	Good for horizontal flow. Tends to underpredict pressure drop and holdup. Recommended by API for wet gas lines.
Beggs & Brill	Use the no-slip option for low holdup. Underpredicts holdup. Most consistent and well-behaved correlation.
Inclined Flow	
Mukherjee-Brill	Recommended for hilly terrain pipelines. New correlation based heavily on in situ flow pattern. Only available model that calculates flow patterns for all flow configurations and uses this information to determine modeling technique.

The correlations available can be divided into pre- and post-computer periods. In the pre-period, the complexity of the correlation was limited by calculational ability. The "fitting" process, therefore, dealt with the overall factors affecting flow only. In the post-period it was convenient to use more variables and introduce more empirical constants. There are advantages in this from a correlation development viewpoint, but it also allows one to out-compute real knowledge about the system being designed. The bottom line of all this is that one should not choose or discard a calculation method because of its age or complexity.

The circumstances affecting a calculation strategy affect the choices. If one works for a large company with research capability, it is likely that test have been run on actual systems. It is equally likely that basic models have been modified to fit this information. An expanded or modified version of existing correlations has been developed. This is good because it adds to knowledge. The correlation may be better than the basic one from which it comes, but it still is not a panacea for all two-phase flow problems. The problem with such in-house correlations is that they sometimes are anointed with "sacred" powers they cannot possess.

PHYSICAL PROPERTIES

Fluid properties effect two-phase flow calculations much more than normally considered. The overall simulation accuracy depends on accurate fluid property calculations. The selection of the most appropriate correlation for any application is one of the more difficult choices made. Unfortunately most engineers avoid this choice and choose the "default" – which should be renamed the "not-my-fault" – option. The proper correlation to use is quite simply the one that matches your system.

Fluid property correlations can be divided into two major categories: 1) non-compositional and 2) compositional models. Non-compositional modeling is an empirical method of estimating those properties: vapor-liquid split, viscosity, etc. Non-compositional models are often employed because the information required – gas specific gravity, liquid specific gravity, GOR – are readily available. These methods are often less accurate at predicting certain critical physical properties. Compositional models require gas and/or liquid analysis which are often not available. Use of these models will not necessarily improve the model. Several things are necessary for these models to improve the accuracy of the fluid flow calculations – proper sampling techniques, proper analysis, proper correlation choice, etc.

Non-Compositional Correlations

Non-compositional fluid models predict bulk fluid properties from the gas and/or liquid specific gravities, gas oil ratio, water oil ratio, etc. The fluid property correlations are often divided into five subcategories: 1) blackoil, 2) gas condensate, 3) single-phase gas, 4) single-phase liquid, and 5) pure component.

Blackoil Model

"Blackoil" refers to a multi-phase fluid model commonly used in the oil and gas industry. This model predicts fluid properties from the specific gravities of the gas and oil, and the volume of gas produced per volume of oil (GOR). Empirical correlations determine the phase split and physical properties are then calculated for each phase.

The blackoil model assumes the liquid at stock tank conditions remains in the liquid phase at all pressures and temperatures. The gas can exist as free gas or as dissolved gas. This type of model is often used in reservoir simulations where the complexity of the fluid flow limits the allowable complexity of the physical property modeling. This type of model should only be used for relatively stable fluids – API less than 45 and GOR less than 5000 scf/Bbl.

In the blackoil model solution gas/oil ratio (SGOR), formation volume factor (FVF) for oil and water, and solution gas/water ratios (SGWR) are required. SGOR and SGWR are the fluid properties which determine the "in-situ" phase split. The in-situ properties, as calculated by the blackoil model, are an approximation of the true vapor-liquid equilibrium (VLE) and physical properties at a particular temperature and pressure. The FVF is used to calculate in-situ liquid densities. Several solution gas/oil ratio correlations have been developed.

Oil formation volume factor (FVF) is the in-situ volume of the oil phase occupied by one standard volume of oil. It is normally expressed as bbl/stb. This factor is used to determine the in-situ liquid density. The total volume of the oil is actually the volume of oil plus the volume of dissolved gas both corrected for pressure.

Solution gas water ratio (SGWR) is the in-situ standard volume of gas dissolved per unit volume of stock tank water. It is normally expressed as scf/stb. A correlation developed by Culberson and McKetta is often used for this calculation. This correlation assumes the dissolved gas is pure methane. The accuracy of this method is normally within 5 percent. The water formation volume factor (FVF) is the in-situ volume of the water phase occupied by one standard volume of water. It is normally expressed as bbl/stb. Water FVF is calculated from water densities. The effect of dissolved water is often neglected in two-phase flow

programs. The effect on flow calculations is minimal but the effect on gas flow from a down stream separator can be significant.

Physical properties must be calculated for each phase. The density calculated greatly effects the prediction of flow by affecting the mixture density, estimated flow regime, etc. as discussed below. Viscosity is the most important, but often neglected parameter in two phase flow. Various correlations are available from different programs. In addition, some options are available for viscosity which allow the oil-water mixture is treated as one fluid, two fluids, or an emulsion. Surface tension is also used in several correlations to predict the flow regime. Once used to predict the flow regime the effect of surface tension is minimal on two-phase flow calculations. However if the wrong flow regime has been predicted, the whole calculation is suspect. In addition, most programs give the option of entering tabular physical property data to replace calculated properties..

Gas Condensate

Most programs that offer a "Gas-Condensate" option using the procedure of API 14B. This model assumes no liquid is present below the lower dew point or above the upper dew point. In-situ condensate flow rate is calculated by multiplying the standard volumetric flow rate by a pseudo-formation volume factor (pseudo-FVF). The pseudo-FVF is a function of condensate specific gravity, dew point pressure, pressure, and temperature. This model assumes the condensate density is constant and equal to the density at stock tank conditions. Dewpoint pressure, mass phase split and surface tensions of condensate and water are calculated from empirical relationships. Dewpoint pressure is a function of condensate specific gravity and temperature.

Single Phase Gas

Single phase gas properties are usually calculated by the gas physical properties listed in the blackoil model. Gas density is calculated from Equation 3.12.

Single Phase Liquid

This model uses the liquid gravity to calculate physical properties. The oil and water correlations listed above are often used with two modifications: the SGOR is zero and the liquid is assumed to be incompressible. Care should be used when entering specific gravity as many programs assume a specific gravity larger than 1.0 (9 °API, for example) is water not oil.

Pure Components

Most flow programs offer pure component options. These are usually limited to water (steam) and CO₂. Tabular or specific data correlations are used. The accuracy of most of these correlations is high for the range of normal operations.

Compositional Fluid Models

Compositional fluid modeling is a method for describing a stream based on its pure components. Equilibrium phase splits and phase properties are determined by blending the properties of the stream constituents. Typically equations of state are used to predict properties and VLE. The accuracy of a compositional fluid model depends on the accuracies of the component properties, the mixing rules, the equation of state, and the accuracy of the sample. For petroleum fractions, the programs must develop pseudo components to use in the correlation. These methods are reviewed in Volume 3 of "Gas Conditioning and Processing.". Compositional models predict the following fluid properties: 1) equilibrium K-values and VLE; 2) gas and liquid enthalpies; 3) gas and liquid densities; 4) gas and liquid viscosities; 5) surface tension; 6) gas and liquid thermal conductivities.

The Calculation Suite

There is no way one can designate in general which suite to use. The problem is very much like that in well logging or PVT behavior. The choice is dictated by what one knows about the system under study.

The newer methods tend to be more general and will be applied on a broader basis than the older methods. It is recommended, though, that one of the older methods be included in each suite. For horizontal flow where the liquid loading is within the correlation and gas is the primary use, the modified Flanigan approach may be one suitable method. If mist flow is indicated, Duns and Ros is a candidate for both horizontal and vertical flow. If liquid is the only continuous phase (bubble flow), a regular liquid flow equation with correction of physical properties for gas bubbles may be used. For vertical bubble flow, Poettmann-Carpenter may be used within the limits specified; outside these limits a modified version like that of Hagedorn-Brown may be employed.

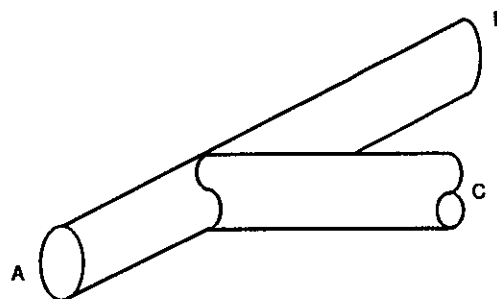
What do you do with the results of the calculations? Regard them as the *range* of pressure performance expected. Design the total system to operate *anywhere* within the pressure range and liquid handling need indicated.

A different suite may be used for early life calculations than at later life full-flow conditions. When operating at or near design capacity, pressure and liquid fluctuations should be reasonably predictable.

FLOW SPLITTING

The flow in a line may be split between two or more lines at some point. This occurs in distribution systems, at surface facilities, river crossings and the like. In a two-phase system the liquid split will not be the same as that of the gas, necessarily.

There are three forces involved in determining the amount of liquid entering each branch of the tee shown in the sketch -- *inertial*, *gravitational* and *centripetal*. If the flow is A to B the inertial forces tend to keep the liquid flowing in a straight line. The gravitational forces tend to pull the liquid to the bottom. Thus, the angle of the side connection should make a difference. Everything else being equal, a tee in a vertically upward position should get less liquid than one in the downward position. Where the side branch is horizontal, gravity should have little effect on the split.



As the gas enters a side branch, C, a circular motion is imparted to it. When a mass traverses a circular path with constant speed, the velocity changes in *direction* but not in *magnitude*; velocity is a vector quantity. If one examines what must go on according to the laws of physics, an unbalanced set of forces occur which cause acceleration toward the center of the circle. It takes energy to produce acceleration so that a localized pressure change (drop) occurs (somewhat akin to that in an orifice meter).

Fluid always flows in a direction toward the lowest pressure available. In a horizontal tee, liquid will only enter the side tee if the centripetal forces produce a driving force in that direction that over comes the inertial forces for straight-ahead flow.

Newton's First Law of Motion says that a mass in motion will move in a straight line unless acted on by unbalanced forces causing it to deviate. In this case, the forces are gravitational and centripetal, the equations for which are:

$$\text{Gravitational: } F = mg/g_c \quad \text{Centripetal: } F = mv^2/r$$

Where: r = radius of circular motion

Centripetal force, a major consideration in flow-splitting, depends on velocity, the radius of the pipe and the mass involved. Velocity, in turn, depends on total flow rate. Theory suggests that the amount of liquid entering a sidebranch would depend on the amount of gas entering that branch. This has been confirmed experimentally.(10.29-31) Figure 10.17 shows the result of a test on a gas system in the Netherlands by Oranje and co-workers.

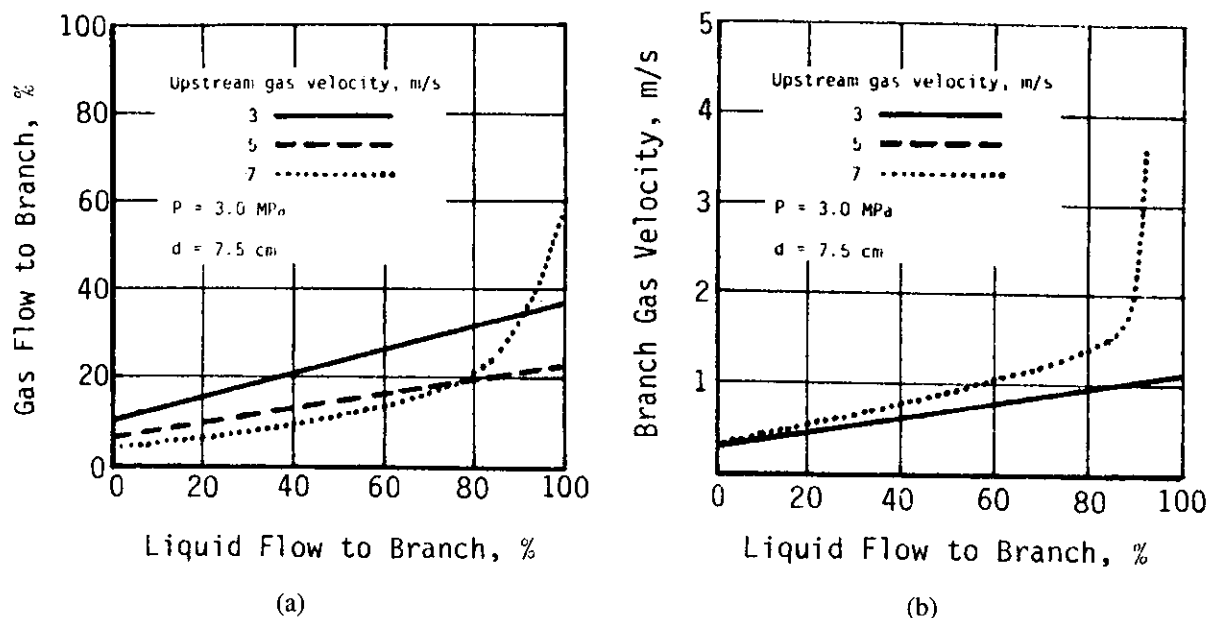


Figure 10.17 Behavior of Two-Phase Grid System

Notice the effect of upstream velocity on behavior. At 5-7 m/s [16-25 ft/sec] velocity the curves are linear; at 3 m/s [10 ft/sec] a break occurs at about 85% of the liquid flowing to the branch. This shows the effect of velocity on behavior.

Hong^(10.31) expanded this kind of study. Although the work was done at low pressures in 9.5 mm [3/8 in.] transparent pipe, the results check reasonably well with Oranje's work in 76 mm [3 in.] pipe at higher pressure.

Once again, as gas velocity increases centripetal force increases. For a given velocity, an increase in the liquid amount increases inertial force without affecting centripetal force. Thus, more of the liquid moves in a straight line. As viscosity increases, all other factors remaining constant, more of the liquid enters the side branch. Why? Increased liquid viscosity causes increased slippage between gas and liquid, decreasing liquid velocity (and thus its inertial forces).

When the gas/liquid flows into "C" and out "A" and "B" the liquid split is proportional to the gas split when 15-85% of the gas enters "A" or "B."

OFFSHORE RISERS

The flow in a riser may differ from that in a wellbore where it is at the end of a relatively long, essentially horizontal, flow line. Holdup and surging from that line is transmitted to the relatively short riser. The riser may have to handle far more liquid than a well because the line can feed it liquid surges that far exceed those possible by gas-lift or reservoir mechanisms.

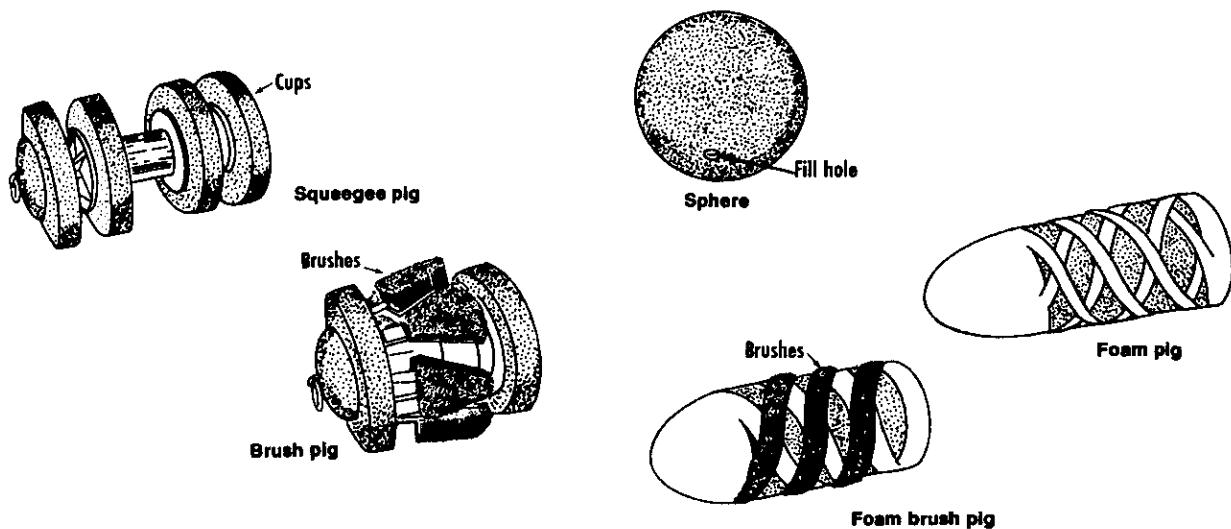
The pressure at the bottom of the riser can vary if the holdup in the riser is not about the same as in the line feeding it. If the riser holdup is too large and the gas velocity too small to provide continuous liquid lift, too much of the liquid reverses and flows downward. Liquid accumulates, causing an unstable pressure situation. This is relieved by large liquid slugs periodically leaving the riser at high velocity. The changes in liquid amount and the corresponding pressure changes can be dramatic. Onshore a large "slug catcher" installation can be provided; offshore this is uneconomical on the platform itself. This is one of the practical reasons why the line immediately ahead of the riser should be horizontal or have a slightly upward slope of 2 to 5 degrees. From some unpublished tests, the length of this section probably should be several times the riser height. The upward incline eliminates a possible "sump" effect and serves to decrease pressure/holdup instabilities.

Many studies of this problem have been made, but most have not been published. Data that have been published show how important gas velocity is in lifting liquid.^(10.32-33) Gas velocities above 2.5-3.0 m/s [8-10 ft/sec] are required to minimize liquid slug length and degree of surging. It is feasible to hold liquid slug size to 10% of the riser volume.

Holding a back-pressure (choking) at the top of the riser can dampen severe slugging. But, this is not a substitute for maintenance of velocity. If flow rate varies widely, the use of multiple risers may be indicated.

PIPELINE PIGGING

Pipeline pigs come in many forms for different applications: (1) removal of liquids collecting in lines during operations or used for hydrostatic testing, (2) to scrape wax, scale and other solids from the pipe walls, (3) to separate different liquids in product pipelines, and (4) to apply internal pipe coatings for corrosion protection.



Guidelines for selecting and using pigs are readily available.^(10.34-37) There are three general classifications: scraper or brush type, foam type and spheres. The choice depends on service and the need to pass through valves and turn corners.

The design and operation of launchers and receivers is summarized in Reference 10.35.

HYDRAULIC DIAMETER

Most fluid flow equations contain a diameter term, expressed as "D" or "d." When the duct for fluid flow is not circular, one must use an equivalent diameter in these equations. It is found as shown below if the duct is running full:

Annulus:	$\left. \begin{array}{l} \text{inner diameter} = d_i \\ \text{outer diameter} = d_o \end{array} \right\}$	$d_e = d_o = d_i$
Square:	side = a	$d_e = a$
Rectangle:	sides a and b	$d_e = \frac{2ab}{a+b}$

The substitution of d_e for d in an equation theoretically is not correct when the ratio of the inner to the outer diameter exceeds 0.3 but appears suitable for use in tubing and casing.

In annular flow the wall friction might vary with the two conduits involved. If a roughness correction is being made, some average is necessary. One approach is to use the relative diameters to prorate roughness effects.

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11

SEPARATION EQUIPMENT

The design of equipment for the separation of vapors and liquids is essential to almost all processes. The design concepts of a simple separator may be extended to several other processes such as fractionation towers, two-phase flow, slug catcher design, desalters, etc. The purpose of this chapter is to review the principles governing the basic separation process and set forth some criteria for use in the planning and operation of the equipment involved.

The basic equipment for separating liquid from vapor uses both *gravitational* and *centrifugal* force. The gravitational force is utilized by reducing velocity so the liquid can settle out in the space provided. Centrifugal force is utilized by changing the direction of flow. A true separator, as defined herein, depends on gravitational force to a substantial degree and has sufficient liquid retention time to allow for effective vapor-liquid disengagement.

FABRICATION SPECIFICATIONS

Most process vessels are fabricated to applicable codes established by government agencies, professional and trade groups, and/or by individual companies. These codes allow specification of fabrication standards in a convenient manner, provide legal protection and help ensure a safe environment for personnel. By specifying acceptable code construction, a company demonstrates that it is operating in a prudent manner.

Current addresses for detailed information about some of the commonly used codes are:

ASME Pressure Vessel Code	American Society of Mechanical Engineers, United Engineering Center 345 East 47th Street, New York, NY 10017 USA
Netherlands	Regels Voor Toestellen onder Druk, Ministry of Social Affairs Government Printing Office, Christoffel Plantijnstraat, The Hague, Netherlands
Germany	A. D. Merkblatt Code published by Carl Heymanns Verlag KG 5 Cologne 1, 18-22, West Germany
Italy	A.N.C.C. published by Casa Editrice Luigi Di G. Pirola, via Comelico 24 (P. O. Box 3680), Milano, Italy (24688)
Sweden	Published by Tryckkarlskommissionen IVA, P. O. Box 5073 S-102 42, Stockholm, Sweden
United Kingdom	BS 5500, British Standards Institutions, London, England
Japan	Dai Isshu Atsuryoku Youki Kouzou Kikaku, Japan Boiler Association 5-35-4 Shiba Minato-ku, Tokyo, Japan
Also	Japanese Standard Association, 4-1-24 Akasaka, Minato-ku, Tokyo, Japan

The ASME Code Division 1 will be emphasized here inasmuch as this is the most commonly used code. A brief discussion will be given on the applicability of other codes and possible design advantages and disadvantages.

Vessel Shell Thickness

The basic formula for calculating the required wall thickness for a vessel under internal pressure is given by the ASME Section VIII, Division 1, Section UG-27, as follows:

$$t = \frac{PR_i}{SE - 0.6P} \quad P = \frac{SEt}{R_i + 0.6t}$$

$$t = \frac{PR_o}{SE + 0.4P} \quad P = \frac{SEt}{R_o - 0.4t}$$
(11.1)

Where: t = shell plate thickness (exclusive of corrosion allowance, in same units as R)
 R_i = internal radius of shell (exclusive of corrosion allowance)
 R_o = external radius of shell
 P = working pressure
 S = maximum allowable stress (If taken as 25% of tensile strength for 483 MPa [70 000 psi] steel, $S = 120$ MPa [17 500 psig].)
 E = joint efficiency, as follows:

Double-Welded Butt Joints		Single-Welded Butt Joints (backing strip left in place)	
Fully radiographed	1.00	Fully radiographed	0.90
Spot radiographed	0.85	Spot radiographed	0.80
No radiograph	0.70	No radiograph	0.65

For spherical shells,

$$t = \frac{PR}{2SE - 0.2P}$$
(11.2)

When seamless pipe is used for vessel shells, one can use Equation 11.1 by substituting the proper tensile strength. It should be noted, though, that seamless pipe wall thicknesses have an allowable variation of 12.5% from the nominal.

Weight and Deck Area Calculation

Equation 11.1 gives the wall thickness for the shell for any vessel under internal pressure. The ASME code has many equations for calculating wall thicknesses of various specialized parts of vessels. In general, Equation 11.1 will give the largest thickness for the vessel and may be used for preliminary weight calculations. The pressure used is the design pressure of the vessel and must be chosen with more care than often is used. Many companies will choose the design pressure to be 1.1 times the normal operating pressure of the vessel. As discussed in the instrumentation section, this often is inadequate if any pressure controls or alarms are required by the design. A more reasonable preliminary estimate of the design pressure is 1.2 times the normal operating pressure.

Although Equation 11.1 is strictly for ASME VIII Division 1, the thickness required for other codes may be estimated by the following rules of thumb. The ASME VIII Division 2 thickness can be taken to be 75% of that calculated by Equation 11.1. The German "codes," British BS-5500, the Japanese pressure vessel code JIS B-8243, Dutch Stoomwezer and most of the European codes may be estimated at 70% of the

thickness of Equation 11.1. These are estimates only and are not valid, particularly at elevated temperatures, cyclic conditions, or large nozzle loadings.

The weight of an empty vessel (including heads) may be estimated from the following equations:

<u>Metric</u>	<u>English</u>	
$W_b = 3.47 \text{ dt}$	$W_b = 15 \text{ dt}$	(11.3)

Where: W_b = mass per unit length, kg/m or lbm/ft
 d = internal diameter, cm or in.
 t = wall thickness, including corrosion allowance, cm or in.

The weight of the internals (W_I) also may be estimated from Table 11.1. The weight of the external nozzles (W_N) also may be estimated from Table 11.1. The weight of the vessel can therefore be estimated as

$$W_v = W_b L + W_I + W_N \quad (11.4)$$

Where: L = S - S length in m or ft

Note that this is the weight of the vessel and does not include fluids from operation or hydrostatic test.

For skidded equipment the following factors have been found satisfactory for preliminary estimates:

Piping W_p - 40% of W_v	Skid steel W_S - 10% of W_v
Electrical & instrument, W_E - 8% of W_v	$W_{\text{Skid}} = W_v + W_p + W_E + W_S$

TABLE 11.1
Weight of Pressure Vessel Accessories

Vessel Internal Weight in Pounds (W_I)Manways				
Vessel Diameter	Mist Eliminators		Distillation Trays	
	Vane	Mist Mat	Normal	Light Wt.
2'0"	14	12	70	50
2'6"	17	15	105	75
3'0"	22	19	160	110
3'6"	28	23	210	150
4'0"	33	27	280	200
4'6"	40	32	350	250
5'0"	46	36	440	310
5'6"	53	41	520	370
6'0"	60	46	625	440
6'6"	68	51	730	515
7'0"	75	56	850	600
7'6"	84	62	970	685
8'0"	93	68	1110	780
8'6"	103	74	1240	875
9'0"	116	80	1400	980
9'6"	126	86	1550	1100
10'0"	137	93	1750	1220
10'6"	143	99	1900	1340

Manways	
ANSI Class	16"
150	525
300	830
400	990
600	1325

External Nozzle Weights in Pounds (W_N)											
ANSI Class	Nozzle Sizes (Inches)										
	2	3	4	6	8	10	12	14	16	18	20
150	10	30	45	65	100	140	185	240	320	345	410
300	15	30	55	95	130	170	245	325	440	565	670
400	20	40	70	100	150	205	295	370	490	580	705
600	20	40	75	120	180	270	330	485	675	825	1020

Care should be used when applying these factors as an excessive number of block valves or optional flow paths may increase the piping weight. For more accurate estimates, a preliminary piping layout should be made for all major piping (4 inch and larger) and the piping weight taken as 150% of the weight take off of the major piping. Experience has shown that as a project progresses the "additional" piping required (simply piping not originally thought of) increases dramatically so that all preliminary estimates must be given a healthy safety factor to account for the additional piping and equipment certain to be "found." The following preliminary estimates for plot area of totally skidded process vessels can be made:

	Horizontal Vessels	Vertical Vessels
Skid Width	I.D. x 2	I.D. x 2
Skid Length	S/S x 1.5	I.D. x 2.5
Skid Height	I.D. x 2 + 1 meter	S/S x 1.5 + 1 meter

Note these are approximations and assume a nominal amount of on-skid piping. If pumps are included on the skid, care should be taken to ensure adequate skirt height is provided for NPSH requirements and additional piping space is provided for bypassing pumps, minimum flow requirements, etc. For better estimates, a preliminary "loose" piping layout should be made. Experience has shown that if a piping layout is "tight" in the preliminary stages it will not fit in the detailed stage. Additional plot plan and deck loading estimates for specific pieces of equipment are given in Appendix 11A.

Vessel Configuration

Most fabricators, as a matter of simple economics, must stock a limited amount and size of shell plate and heads. Furthermore, the major fabrication cost is often labor. Having to cut and chamfer shell plate to produce an "odd-length" vessel may be more costly than to "give away" extra plate. It also might be less expensive to use nominal 3.5 MPa [500 psig] steel in a vessel only needing a 2.1 MPa [300 psig] working pressure if the latter required a special order and handling. The moral: It often is unrealistic to hold the fabricator to an exact length or wall thickness specification; both delivery and cost could be affected. Many fabricators, for example, stock shell plate in 76 cm [2.5 ft] increments. Head seam to head seam (S-S) dimensions for the increment stocked is the most economical. One should establish minimum or maximum dimensions, but some latitude should be given the vendor since it may affect both price and delivery time. Standard practice should be to insist that the fabricator quote to your minimum sizes and then offer an alternate quotation for consideration if he desires.

VAPOR-LIQUID SEPARATION EQUIPMENT

Separation of oil and gas is a critical field processing operation. As producing pressures have risen and lighter condensates are produced, efficient separation has become more critical than ever. Moreover, some of the new concepts have been applied to advantage on old lease producing oil at moderate pressures.

As gas transmission lines raise their standards, separation becomes a part of the overall field processing necessary properly to condition the gas. Consumer specifications are becoming more rigorous.

A "complete" separator must have the following:

1. A primary separation section to remove the bulk of the liquid from the gas
2. Sufficient liquid capacity to handle surges of liquid from the line
3. Sufficient length or height to allow the small droplets to settle out by gravity (to prevent undue entrainment)
4. A means of reducing turbulence in the main body of the separator so that proper settling may take place
5. A mist extractor to capture entrained droplets or those too small to settle by gravity
6. Proper back-pressure and liquid-level controls

All separation may be divided into two classes – scrubbers and separators. This classification is loose, and is only to distinguish that equipment having all the above features from that which does not. In this nomenclature, a scrubber is any device designed for separation of liquid from gas that does not have feature (2) and possibly (6). It is a vessel, then, designed to handle relatively small quantities of liquid with no large degree of surging.

Choice of the separation device obviously hinges on economics. By virtue of its smaller size and simplicity, a scrubber is cheaper in initial cost and easier to install. Use of scrubbers should be carefully examined to see what ultimate effects liquid carryover might have on operations downstream. Such an examination often reveals that the slight additional cost of a properly sized separator is indeed very cheap insurance.

It is poor practice to use a scrubber as the primary separation means at the well. Production of liquid is often unsteady and there is a definite need for liquid surge capacity. Even dry gas wells often will start producing liquid, including salt water, during their later life. Consequently, scrubbers are recommended only for:

1. Secondary separation to remove carryover fluids from process equipment such as absorbers and liquid dust scrubbers
2. Gas line separation downstream from a separator and where flow lines are not long
3. Miscellaneous separation where the gas-liquid ratio is extremely large

Separator Components

Figures 11.1 and 11.2 are typical cutaway views showing the various separator parts. Primary separation is normally accomplished with the aid of centrifugal force. Either a tangential inlet or internal diverter is used to cause a spinning motion of the incoming gas. The latter is preferred by most, since it permits in-line piping. With this approach, the large liquid droplets are thrown out of the main gas stream, which limits entrainment.

All standard separators' published capacities are based on steady flow. However, experience dictates that surge capacity be built in. There is no way of determining the necessary surge capacity, as the degree of surging can never be known exactly in advance. Consequently, the amount of capacity chosen has to be a compromise between cost and the average process requirements. Standard size separators, produced by reputable manufacturers, have very similar dimensions; and all have, in effect, incorporated an arbitrary amount of surge capacity.

The length of a horizontal separator has a greater effect on capacity than the height of a vertical type. In the horizontal vessel the path of any droplet ideally has a trajectory similar to that of a shell from a gun. Therefore, the length necessary depends on:

- | | |
|--------------------|-------------------------|
| 1. Droplet size | 4. Vessel diameter |
| 2. Gas velocity | 5. Degree of turbulence |
| 3. Droplet density | 6. Gas density |

In the ideal case, turbulence may be neglected and effect of gravity assumed constant. Droplet velocity affects length – as it increases the droplets will travel farther before settling out. Although corrections have to be made for the nonideal conditions that exist, an analysis shows that the capacity of horizontal separators of a given diameter may be increased by increasing length.

Application of the same principles to vertical separators shows slightly different results. There, the droplet velocity is working against gravity, which makes separation more difficult. In order to keep separator height reasonable, velocities must be smaller. Sufficient length is needed only for the velocity to become zero and for the droplet to start falling. Increasing the length above this point accomplishes little good.

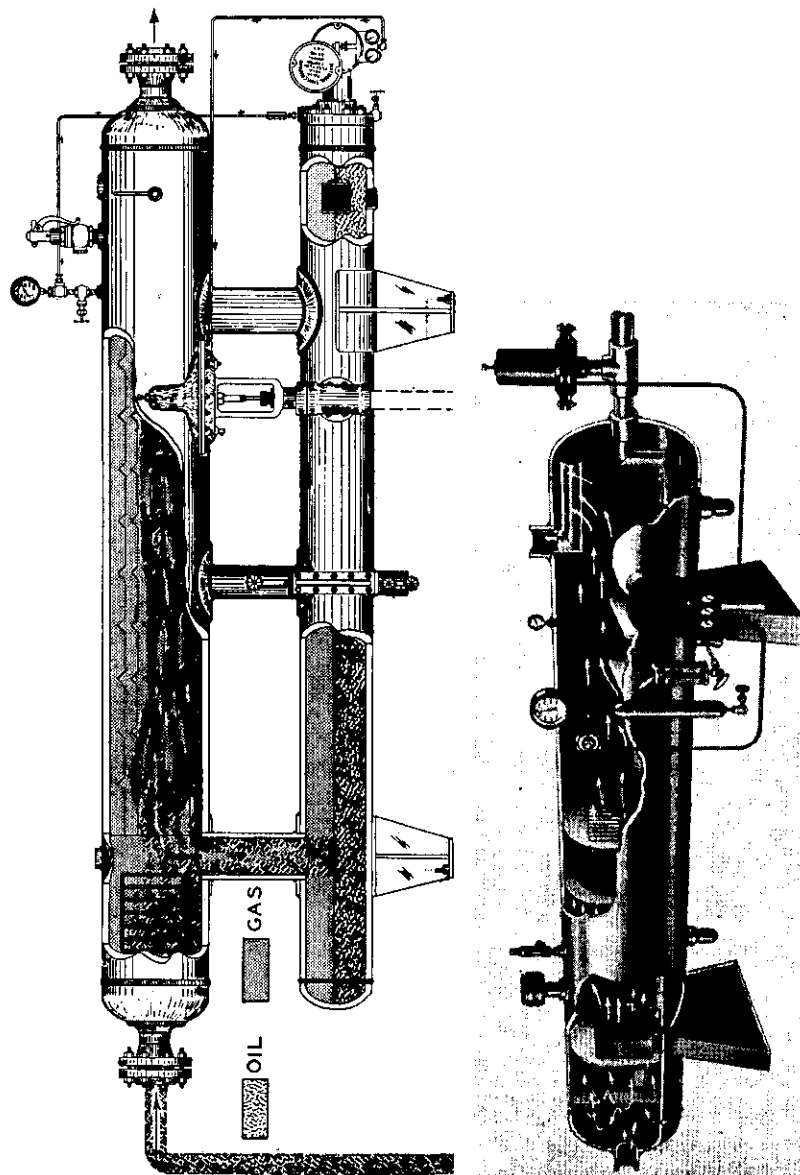


Figure 11.1 Cut-Away View of Horizontal Separators

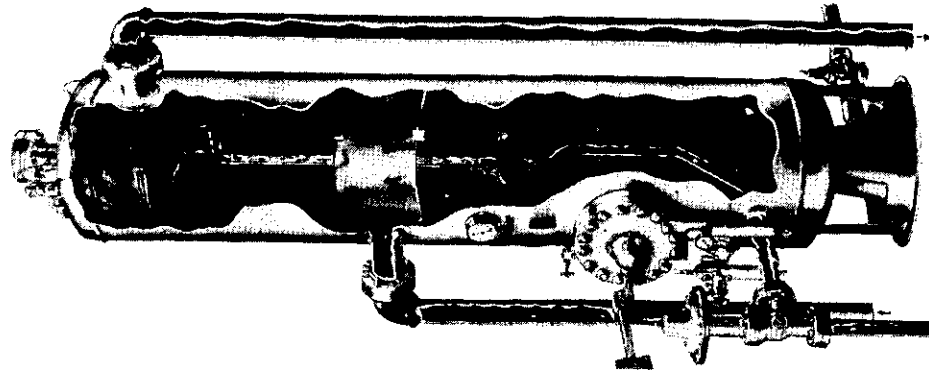


Figure 11.2 Cut-Away View of a Vertical Separator

If too much liquid is carried into the separation section, the falling of the small droplets is hindered and entrainment becomes more prevalent. This is one reason internal design is critical. Theory does not give the designer sufficient tools, so design is best accomplished by actual test.

The mist extractor is designed to trap those droplets that have for any reason been carried beyond the separation section proper. This subject will be explored more thoroughly in a later section.

Principles of Separation

Separation equipment employs one or more of the following mechanisms: (1) gravity settling, (2) centrifugal force, (3) impingement, (4) electrostatic precipitation, (5) sonic precipitation, (6) filtration, (7) adhesive separation, (8) adsorption, or (9) thermal.

The problem is amplified because particles of varying size and characteristics, both liquid and solid, must be removed. Furthermore, size and cost of the equipment required are always practical factors.

Particle Size. This is normally defined by its diameter in microns, $1\mu = 1 \times 10^{-6}$ meters. The particles greater than 10μ may be separated by properly designed equipment. Those smaller, sometimes referred to as particulates, represent a serious problem. Up until the present time at least, the use of settling, impingement and centrifugal force have not been generally effective in handling these small particles. The other processes as yet have not been generally used on high pressure gas because of several serious limitations.

There are several ways of describing the average particle diameter:

1. Diameter of a sphere having average volume:

$$D_p = \left[\frac{\sum(nD^3)}{\sum n} \right]^{1/3}$$

2. Diameter of sphere having average area:

$$D_p = \frac{\sum(nD^2)}{\sum n}^{1/2}$$

3. Diameter of sphere having geometric mean diameter:

$$D_p = c^{-B}$$

Where: $B = \sum(n \ln D) / \sum n$

D_p = average particle diameter

D = diameter of given size particle

n = number of particles of given size (dimensionless)

(D and D_p in consistent units)

The usual probability of particle sizes normally takes the form shown in Curve a, Figure 11.3. Often the curve will be skewed to the right or left as with Curve b, or several peaks may be encountered as with c.

Figure 11.4 gives examples of particle sizes for comparison purposes. The general break point between fog and mist is approximately 50 microns.

The determination of average particle size is relatively easy with solids, but no method yielding reproducible data has yet been devised for liquids. This determination of particle size may be considered somewhat academic because size changes continually in an actual line -- yet it is by necessity the logical starting point in design.

Most particles have no electrical charges of any magnitude; therefore, if a charge is desired (electrostatic precipitation), this normally must be imposed by artificial means. For all practical purposes the particles may be considered inert.

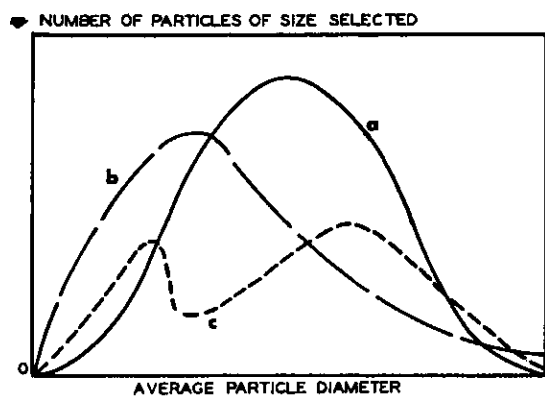
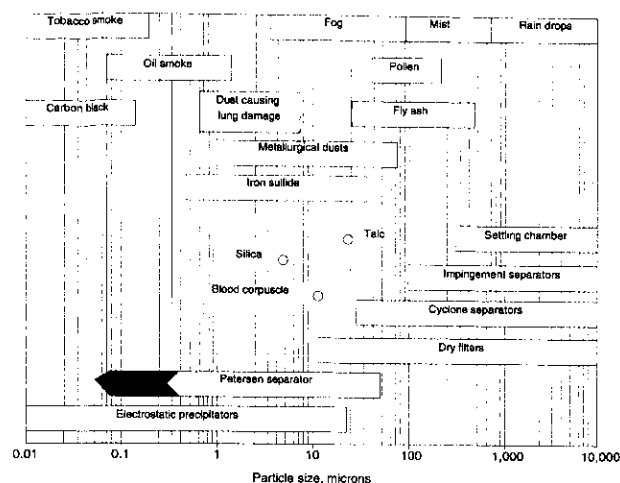


Figure 11.3 Typical Particle Distribution

Figure 11.4 Particle Size Examples^(11.1)

Although particles of widely varying absolute density are encountered, their bulk density is generally not much greater than that of the gas. Consequently, separation by gravity is logically limited to particles of relatively large size. It is only above $10\ \mu$ that their higher density starts having an effect.

Many mist eliminator designs are essentially redistributors of the particle distribution curve. An ordinary horizontal separator might have the particle distributions throughout its length as shown in Figure 11.5.

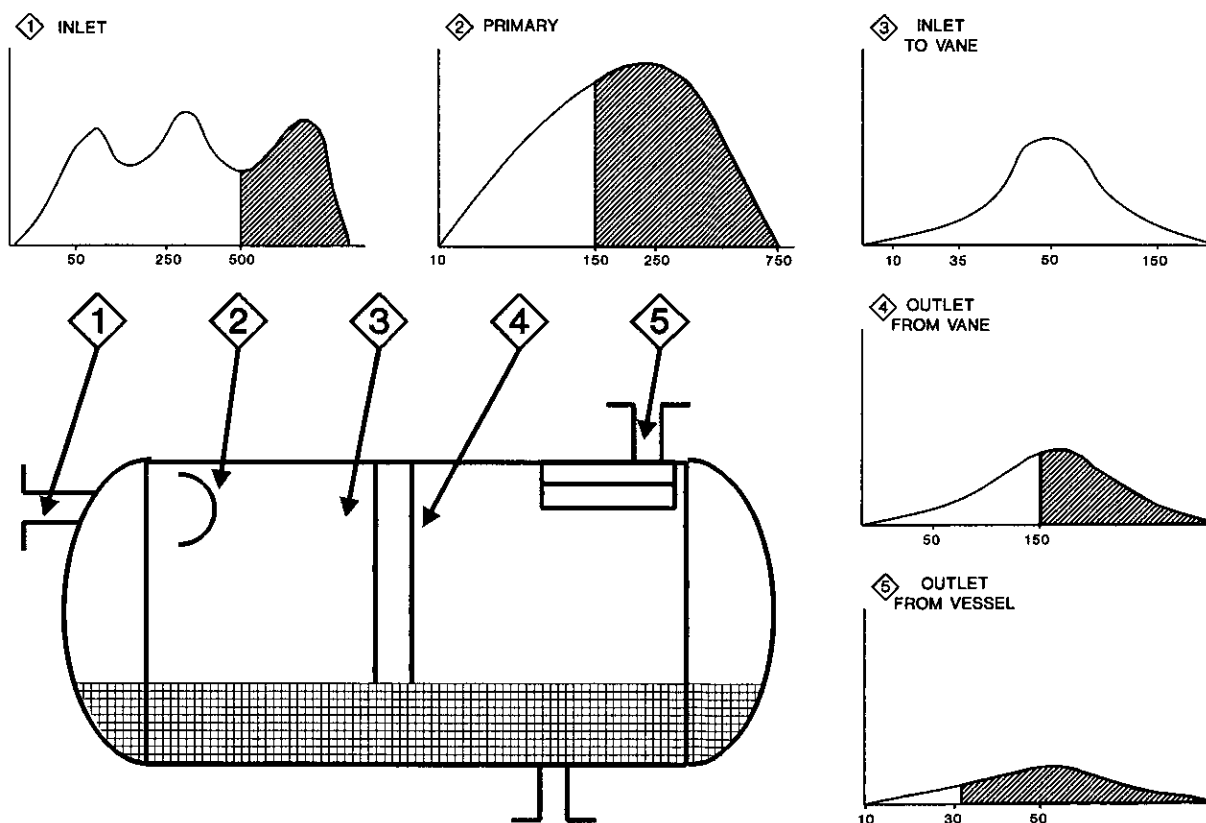


Figure 11.5 Example of Particle Distribution in Separator

The distribution depends on internal design, the character of the fluids involved and the equipment immediately upstream. Taking a large pressure drop across a valve, particularly with high gas-oil ratio condensates, can atomize the liquid and cause small particle sizes. What all of this means is that a given separator's performance depends, to some degree, on the system within which it is placed.

- A. An inlet particle distribution that is skewed and with several distinct particle sizes enters the vessel as shown in (1).
- B. The stream hits the inlet momentum device where bulk liquid (particles larger than 500 microns) are removed.
- C. The distribution shown in (2) enters the primary separation area where gravity effectively separates all particles 150 microns and larger.
- D. The inlet to the vane mist eliminator should have been changed to the distribution in (3). At this point the "vane" type mist eliminator through inertial force, centrifugal force, and reduction of turbulence, coalesces the small droplets into larger droplets. Some of these droplets would be large enough to fall to the liquid by paths provided in the vane section. A significant number of large droplets would pass through the vane as shown in (4).
- E. The outlet from the vane would enter the secondary separation section where gravity would again separate all those particles larger than 150 microns.
- F. This new particle distribution would enter the mist eliminator where an action similar to Step D would occur so that the final outlet from the vessel would have a distribution of particles 30 microns and less.

There are several points to be made about the above description. Many vendors would claim that they remove all the particles down to 10 microns. This is not commercially viable in an entrainment type separator. In oil and gas production, most separators are sized to remove liquid droplets larger than about 150 μm by gravity settling. Smaller droplets must coalesce to form larger droplets ($D_p > 150 \mu\text{m}$) which may be separated by gravity. Mist extractors are commonly used to provide coalescence. Under ideal conditions, a 100 mm [4 inch] thick wire mesh demister pad will allow 30-40% of the droplets larger than 20 μm to pass through, although the efficiency rises to near 100% for droplets larger than 60-70 μm . As you can see, separation efficiency is dependent on particle size distribution.

Experience has indicated that minimum particle size is related to inlet velocity, number of restrictions and amount of pressure drop in the inlet piping. These facts give inlet nozzle guidelines as discussed later, but the vendor usually has no control over the inlet piping. Therefore, although performance guarantees are nice to have, the ultimate responsibility for a system is with the design engineer and not the vendor. Numerous mechanical details will enhance the operation of vessels and should be looked for in the purchase of equipment. These will be discussed briefly later.

Gravity Separation

The terminal velocity of a particle falling through a fluid by the pull of gravity is

$$v_t = \left[\frac{4 g D_p^{N+1} (\rho_p - \rho_f)}{3 A \mu^N \rho_f^{(1-N)}} \right] \left(\frac{1}{2-N} \right) \quad (11.5)$$

Where:

- g = gravitational acceleration
- D_p = particle diameter
- ρ_p = particle density
- ρ_f = fluid density
- μ = fluid viscosity

"A" and "N" are constants related to the flow regime and drag coefficient on the system as determined by

$$K = D_p \left[\frac{g \rho_f (\rho_p - \rho_f)}{\mu^2} \right]^{1/3}$$

The values of "A" and "N" for use with Equation 11.5 are found from the table

Law	K	A	N
Stokes	$K < 3.3$	24.0	1.0
Intermediate	$3.3 < K < 43.6$	18.5	0.6
Newton's	$K > 43.6$	0.44	0

When liquid is being separated from gas in laminar flow (Stokes Law), Equation 11.5 becomes

$$v_t = \frac{g D_p^2 (\rho_L - \rho_g)}{18 \mu} \quad (11.6)$$

Centrifugal Settling. Stokes' Law may be applied to this process if the effect of gravity, g , is replaced by a , the acceleration due to centrifugal force. The terminal velocity then becomes

$$v_t = \frac{a D_p^2 (\rho_L - \rho_g)}{18 \mu} \quad (11.7)$$

By considering the gas velocity to be constant, it is possible to obtain

$$D_p = \left[\frac{9 \mu s}{\pi v_o N (\rho_L - \rho_g)} \right]^{1/2} \quad (11.8)$$

Where: D_p = particle size that will be collected by centrifugal force

Impingement. As the gas approaches a fiber, it curves around but the momentum of the entrained droplet tends to move it straight ahead. It therefore becomes critical to determine the distance across the flow path necessary to stop the particle. On the assumption that Stokes' Law applies:

$$s = \frac{D_p^2 \rho_g v_l}{18 \mu} \quad (11.9)$$

Where:

- a = acceleration due to centrifugal force
- D_p = average particle diameter
- g = acceleration due to gravity
- n = number of particles of given size
- N = "effective" number of turns made in centrifugal separator
- v_t = particle terminal velocity
- v_o = average velocity of particle laden gas
- v_l = initial velocity of particle
- ρ_L = density of liquid
- ρ_g = density of gas
- μ = viscosity of gas
- s = radial path of particle
- π = 3.1416

Metric	English
m/s	ft/s
m	ft
9.81 m/s^2	32.2 ft/s^2
m/s	ft/s
m/s	ft/s
m/s	ft/s
kg/m ³	lbm/ft ³
kg/m ³	lbm/ft ³
kg/m·s	lbm/ft·s
m	ft

SEPARATOR SIZING

The determination of separation vessel diameter is based on the above principles. It is a semi-empirical approach since one cannot measure things like droplet size and other variables. Furthermore, many of the assumptions in the above equations are not satisfied in actual practice.

Consequently, one form of equation used is

$$v = K_s \left[\frac{\rho_L - \rho_g}{\rho_g} \right]^{0.5} \quad (11.10)$$

Where: ρ_L = liquid density
 ρ_g = gas density
 v = allowable gas velocity
 K_s = an empirical constant

Metric	English
kg/m ³	lb/ft ³
kg/m ³	lb/ft ³
m/s	ft/s
m/s	ft/s

All density terms are at the pressure and temperature of separation.

The value of K_s depends on all factors that affect separation other than density – vortex action, foaming, pulsating flow, liquid flowing in heads, presence of solids, degree of separation needed, separation length, varying gas-liquid ratios, and the like. It is not surprising that K_s varies widely in different applications. How do you predict it? From experience!

The following values of K_s are taken from API 12J.

Separator Type	Height or Length, m [ft]	K_s Factor	
		Metric	English
Vertical	1.5 [5]	0.037-0.073	0.12-0.24
Vertical	3.0 [10], or taller	0.055-0.107	0.18-0.35
Horizontal	3.0 [10]	0.122-0.152	0.40-0.50
	Other	$K_3(L/3)^{0.56}$	$K_{10}(L/10)^{0.56}$
Spherical	All	0.061-0.107	0.20-0.35

Notice that some of the values of K_s were cited as a function of length. The value does vary with length (to a point) but one must use such relationships with caution. For example, most 5-foot vertical separators are used on crude oil streams where the liquid is in large "blobs" with virtually no small droplets. The values shown reflect performance of that kind of stream and not one where the distribution of particle sizes is more symmetrical as shown previously. In some service a 5-foot separator is basically unsatisfactory; K_s is effectively zero.

The length correction for horizontal separators also must be used carefully. The values shown were obtained on separators with a length-to-diameter ratio averaging about 5:1. In effect, K_s only increases significantly with length as this L/D increases. A separator 10 ft by 50 ft may not have higher value of K_s than one 4 ft by 20 ft (the same L/D ratio). The former may tend to possess a slightly higher value of K_s but it would not be as much greater as the correlation would indicate. Unless L/D is greater than 5:1, it is difficult to justify a value of K_s greater than 0.15 [0.50] regardless of length.

If a horizontal separator is to be used with an L/D other than 5, multiply the value of K_s by the term

$$\left[\frac{L/D}{5} \right]^{0.56}$$

to correct for extra length in the absence of foaming. If severe foaming is expected you may wish to use a minimum value of K_s and add extra length, based on experience.

The "v" found in Equation 11.10 is the apparent velocity in the space open to gas flow. In a single barrel, horizontal separator one-half full of liquid, only one half of the cross-sectional area is available for gas flow.

One can convert Equation 11.10 to volume rate of flow by remembering that

$$q_a = (\pi/4)(d^2)(v)(F)$$

Where: q_a = Actual flow, m^3/s [ft^3/s]
 F = fraction of the separator cross-section area available for gas flow.
 The value of "F" is affected by internals as well as liquid.

It is frequently desired to calculate the gas capacity of the separator in terms of volumes measured at standard conditions.

$$q_s = 67\,824 (K_s)(d)^2(F)\left(\frac{1}{z}\right)\left(\frac{P}{P_s}\right)\left(\frac{T_s}{T}\right)\left[\frac{\rho_L - \rho_g}{\rho_g}\right]^{0.5} \quad (11.11)$$

Where: q_s = gas flow rate at P_s and T_s
 d = total internal diameter of separator
 P = separation pressure
 T = absolute separation temperature
 ρ_L = liquid density
 ρ_g = gas density
 P_s = pressure base
 T_s = temperature base
 z = compressibility factor
 F = fraction of total area available to gas

Metric	English
m^3/d	ft^3/d
m	ft
kPa	psia
K	$^{\circ}R$
kg/m^3	lbm/ft^3
kg/m^3	lbm/ft^3
kPa	psia
K	$^{\circ}R$
	dimensionless
	dimensionless

If one wishes to calculate actual volume rate rather than standard volume rate, (P/P_s) , (T_s/T) and z are deleted from Equation 11.11.

Application of Equations 11.10 and 11.11. The allowable "q" for a given diameter and fluid conditions depends primarily on the choice of K_s . The API sets forth a range of possible values. But, this range is large, varying in some cases by a factor of two. What is a suitable value within this range?

In the final analysis the choice must be made from experience and data from comparable separation facilities and the efficiency of separation required. In some field applications, a nominal amount of carryover may be tolerable and a larger K_s (smaller diameter separator) may be specified than in some process applications. Separation equipment ahead of dehydrators, sour gas treaters, compressors, etc. is very critical. Carryover adversely affects performance.

Surging flow is a problem. The gas flow rate used to determine diameter should be the maximum one converted to a daily rate, not the average daily rate. Unless this is done, there may be carryover during each gas surge as actual rate exceeds design rate. Provision must be made also for liquid surge, as discussed in a later section.

Foaming and emulsified liquids also affect capacity. Said foam must be broken to obtain a good vapor-liquid separation. It takes contact surface and time (length) to break it physically. Chemicals like silicone compounds may be used also but they are expensive. In some instances, horizontal separators with an L/D greater than 10:1 have been used to handle foam successfully.

The severity of the foaming problem is exacerbated by the design of the control system ahead of the separator when a substantial pressure loss is involved. When it is known that the liquid tends to foam, this should be considered.

Use of Mass Flow Rate

Oftentimes it is more convenient to use mass flow rate for sizing purposes. The mass velocity (w), in mass per unit area per hour is related to linear velocity by the equation

$$w = 3600 v \rho_g$$

If this relationship is substituted into Equation 11.10 the result is

$$w = 3600 K_s [(\rho_L - \rho_g)(\rho_g)]^{0.5} \quad (11.12)$$

$$\text{Also,, } m = 0.785 w d^2 F \quad (11.13)$$

Where:

- w = mass velocity
- m = mass flow
- v = linear velocity
- ρ_g = gas density (actual)
- ρ_L = liquid density
- d = separator I.D.
- F = fraction of area available to gas
- K_s = gas sizing constant

Metric	English
kg/h m^2	lbm/h ft^2
kg/h	lbm/h
m/s	ft/s
kg/m^3	lbm/ft^3
kg/m^3	lbm/ft^3
m	ft

Combining Equations 11.12 and 11.13 enables one to solve diameter

$$d = \frac{0.0188 \left(\frac{m}{FK_s} \right)^{0.5}}{[(\rho_L - \rho_g)(\rho_g)]^{0.25}} \quad (11.14)$$

The value of "m" is related to volume rate of flow as follows:

Metric (q_{sc} at 15°C and 100 kPa, std m^3/day):

$$m = 0.050 (\text{std m}^3/\text{day})(\gamma \text{ of gas}) \quad (11.15)$$

$$\text{or } m = 0.00173 (\text{std m}^3/\text{day})(\text{MW gas})$$

English (q_{sc} at 60°F and 14.7 psia, std cu ft/day):

$$m = 0.0032 (\text{scf/day})(\gamma \text{ of gas}) \quad (11.16)$$

$$\text{or } m = 0.00011 (\text{scf/day})(\text{MW gas})$$

In the sizing of vertical vapor-liquid vessels like absorbers and fractionators, it is customary to use values of $K_s = 0.043\text{--}0.067 \text{ m/s}$ [$0.14\text{--}0.22 \text{ ft/s}$]. This is designed to give a rather efficient vapor-liquid separation over a short distance of a few feet. This is consistent with short separator experience.

Separator carryover is the origin of a host of operating problems. This is a good vessel with which to adopt a conservative posture. It is relatively inexpensive.

Example 11.1: Calculate the diameter of the vertical separator needed for a gas flow rate of one million std m³ per day if gas density is 80 kg/m³ and the liquid density is 800 kg/m³. Use a value of K_s equal to 0.0508 m/s. The gas relative density is 0.7.

From Equation 11.15

$$m = (0.050)(1\,000\,000)(0.7) = 35\,000 \text{ kg/h}$$

or

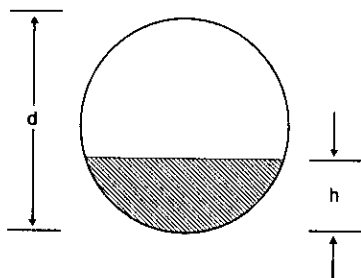
$$m = (0.001\,73)(1\,000\,000)(0.7)(28.96) = 35\,000 \text{ kg/h}$$

From Equation 11.14

$$d = \frac{0.0188 \left(\frac{35\,000}{(1.0)(0.0508)} \right)^{0.5}}{[(800 - 80)(80)]^{0.25}} = 1.01 \text{ m}$$

Values of "F." – The fraction of the total area available for gas flow is shown in Table 11.2. One can extrapolate this table beyond 0.55. The value for $h/d = 0.60$ would be one minus the F value at $h/d = 0.4$. Or, $F = 1 - 0.626 = 0.374$. Appendix 11A contains a figure for the same purpose.

TABLE 11.2
Values of "F" as a Function of Liquid Depth in a Horizontal Separator



h/d	F	h/d	F
0.00	1.000	0.30	0.748
0.05	0.981	0.35	0.688
0.10	0.948	0.40	0.626
0.15	0.906	0.45	0.564
0.20	0.858	0.50	0.500
0.25	0.804	0.55	0.436

Liquid Retention Time

One factor not covered by the gas sizing equations is liquid retention time. It takes a finite time for gas to break out of liquid.

Retention time is an indirect way of fixing the volume of a separator that is necessary to handle the liquid flow rate. Separator liquid volume equals flow rate of liquid times retention time. For a given separator, liquid volume requirements (via retention time) may have a greater effect on size than gas flow rate. This is true particularly of large crude oil separators where the gas-oil ratio is low. The liquid is the controlling factor.

The equation for calculating settling time is

$$V_L = \frac{(q_L)(t)}{1440} \quad (11.17)$$

Where: V_L = required separator liquid section, m³ [bbl]
 t = design residence time, min
 q_L = liquid throughput, m³/d [bbl/d]

Typical retention times are as follows:

Natural gas-oil	1-3 minutes
Lean oil surge tanks	10-15 minutes
Fractionation feed surge tanks	8-15 minutes
Refrigerant surge tanks	4-7 minutes
Refrigerant economizers	2-3 minutes

API 12J gives the following guidelines for gas-oil separation.

Oil Relative Density	Minutes
Below 0.85	1
0.85-0.93	1 to 2
0.93-1.0	2 to 4

Residence time is affected also by composition, foaming, the presence of solids and emulsions.

Retention time is a very important factor in separation vessels where a chemical reaction might be occurring. A good example is water deoxygenation. A retention time of 5 minutes is preferred, with 3 minutes being marginal. This time is necessary for the chemical reaction to proceed and to obtain good chemical mixing.

As noted later in this chapter, some liquid volume must be provided for control reaction time when alarm and shut-down controls are involved.

THREE-PHASE OR LIQUID-LIQUID SEPARATIONS

Three-phase separators handle gas plus two immiscible liquid phases. These two liquid phases might be oil and water, glycol and oil, etc. The best potential application of three-phase separators occurs where space is a major consideration. Do not anticipate perfect separation between the liquid phases, particularly if the gas-oil ratio is high or the oil has any emulsifying or foaming tendencies. Carryover of 2% oil into water or water into oil is not uncommon for three-phase separators operating at design conditions.

Equipment used downstream of a three-phase separator or free water knockout to meet crude and sales specifications or water discharge regulations include:

1. Wash tanks and settling tanks
2. Gun barrels
3. Heater treaters
4. Coalescers
5. Electrostatic precipitators

C. Hydrocyclones

The equipment actually selected depends on several factors, including crude oil gravity and viscosity, water cut, stability of emulsion, specifications, etc.

A three-phase separator must be sized to provide effective vapor/liquid separation and also to provide adequate liquid/liquid separation. Because of the difficulty in separating liquid phases, this criteria often controls the separator size.

Proper sizing of the liquid/liquid separation chamber requires

1. Adequate retention time to reduce turbulence, provide for coalescence, and allow chemical treatment (e.g., demulsifiers and foam inhibitors, etc.) to act.
2. Sufficient cross-sectional area to allow water (or glycol) droplets to settle out by gravity.

Retention Time

The usual approach in design is to allow equal retention time for the two liquid phases. The following are recommendations from API 12J.

Oil Relative Density		Retention Time, min
Below	0.85	3-5
Above	0.85 >100°F	5-10
	80-100°F	10-20
	60-80°F	20-30

For glycol/hydrocarbon separation a minimum retention time of 30 minutes is recommended.

Settling Velocity

The basic settling characteristics of two immiscible phases are represented by Stokes Law (Equation 11.6). Typically, a value of $D_p = 200\text{-}300\text{ }\mu\text{m}$ is suitable. Stokes Law assumes unhindered settling which seldom, if ever, occurs in practice. Stokes Law provides a reasonable estimate of settling velocity when the dispersed phase is dilute (less than 2-3%) in relation to the continuous phase.

Alternative Approach for Settling Velocity

Many people use empirical correlations based on observed data for laboratory or field tests to establish settling velocities for liquid/liquid separation.

Based on laboratory studies of oil-water settling, the correlation shown below has been developed by the author.

$$v = \frac{(A)(C)(\rho_w - \rho_o)}{\mu} (L_c) \quad (11.18)$$

Where:

v = settling velocity

ρ_w = water density

ρ_o = oil density

μ = oil viscosity

L_c = length correction
where L =

A = coefficient

C = correlation constant found from Figure -
11.6

Metric	English
m/h	ft/hr
kg/m ³	lb/ft ³
kg/m ³	lb/ft ³
kg/m·s	lb/ft·s
$0.52(L)^{0.2}$	$1.35(L)^{0.2}$
m	ft
600	6440

The L_c in Equation 11.18 is the effective height of the liquid separation section.

Viscosity is often measured in centipoise (cp). For use in Equation 11.18, remember that

$$1\text{ cp} = 0.000672\text{ lb/ft·s} = 0.001\text{ kg/m·s}$$

Even with large amounts of water, the settling velocity seldom exceeds 6 m/h [20 ft/h] for relatively fresh water. The results shown in Equation 11.18 are usually safe. Actually the rate depends on temperature and whether the two liquids are emulsified to some degree or not (droplet size). The correlation for constant "C" is based on actual tests where individual drops could be observed; there was no emulsion that required heat or chemical to coalesce the water before settling. In addition, it assumes turbulence is negligible and there is no surging.

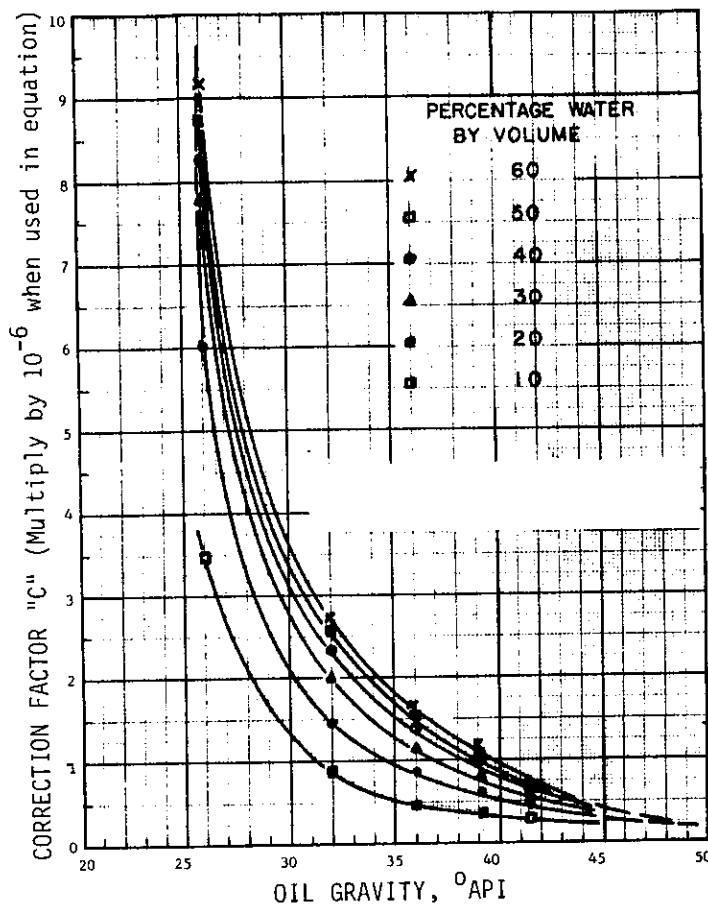


Figure 11.6 Correction Factor for Liquid-Liquid Separation.

Example 11.2: Calculate the settling velocity for a 39°API oil (Sp Gr = 0.83) having a viscosity of 6 cp if the entering stream contains 10% water by volume with a density of 1.0 g/cm³. The separation section height is 0.31 m [1 ft]. From Figure 11.6 the value of "C" is 0.35×10^{-6} .

METRIC:

$$L_c = 0.52 (0.31)^{0.2} = 0.41$$

$$v = \frac{(600)(0.35 \times 10^{-6})(1000 - 830)}{(6)(0.001)} (0.41) = 2.44 \text{ m/h}$$

ENGLISH:

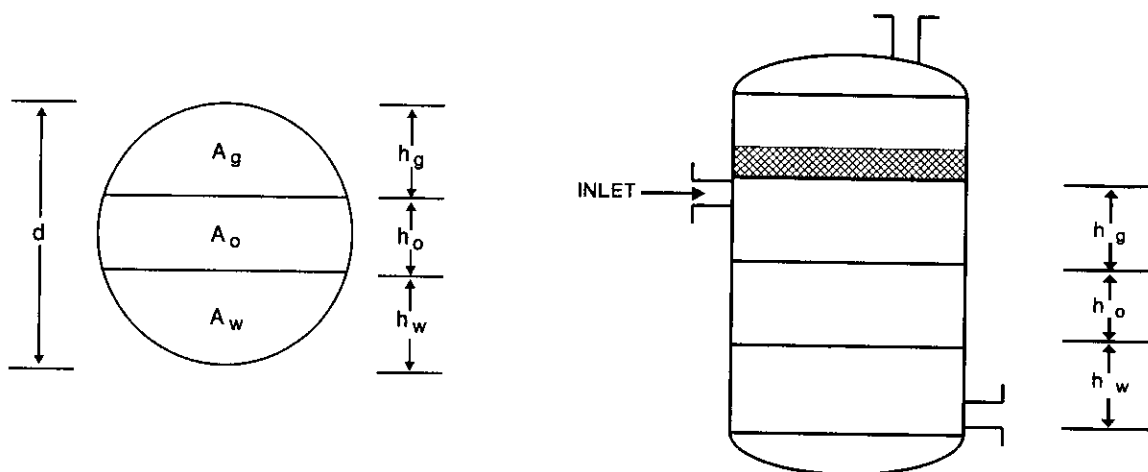
$$L_c = (1.35)(1.0)^{0.2} = 1.35$$

$$v = \frac{(6440)(0.35 \times 10^{-6})(62.4 - 51.8)}{(6)(0.000672)} (1.35) = 8.00 \text{ ft/h}$$

The velocity predicted by Stokes Law may be 2-3 times that observed. But remember, Stokes Law assumes unhindered settling. Notice in Figure 11.6 that the settling velocity decreases with amount of water. This is an indirect indication of the effect of hindrance.

The actual size of the separation section must meet both the retention time and settling velocity criteria. Several vessel geometries can be used. For three-phase separation, horizontal separators are generally preferred because increasing the vessel length decreases settling velocity. Length-to-diameter ratios typically vary from 3 to 5.

A series of equations can be written for gas-oil-water separation using the nomenclature shown in the figures below.



$$t_o = \frac{h_w}{60 v_o} \quad \text{and} \quad t_w = \frac{h_o}{60 v_w} \quad (11.19)$$

Where:

- h_o = height of oil, m [ft]
- h_w = height of water, m [ft]
- v_o = terminal velocity of oil from Equation 11.6 or 11.18
- v_w = terminal velocity of water from Equation 11.6
- t_o = residence time of oil, min
- t_w = residence time of water, min

The length required can be calculated by the equation:

$$L_o = \frac{t_o q_o}{A_o} \quad \text{and} \quad L_w = \frac{t_w q_w}{A_w} \quad (11.20)$$

Where:

- L_o = length req'd by oil flow, m [ft]
- L_w = length req'd by water flow, m [ft]
- q_o = rate of oil flow, m^3/min [ft^3/min]
- q_w = rate of water flow, m^3/min [ft^3/min]

The area occupied by each liquid phase in a horizontal separator can be calculated from Table 11.2, or from a figure in Appendix 11A, once values of h have been fixed. Of course, for vertical vessels the areas should be equal.

VESSEL INTERNALS

The proper selection of internals can enhance significantly the operation of separators. Proprietary internals often are helpful in reducing liquid carryover at design conditions. But, they cannot overcome a basically improper design or operation at off-design conditions.

Mist Extraction

Production equipment involving the separation of oil and gas often uses *impingement-type*, mist-extraction elements. This element is usually of the vane type or of knitted wire.

The *vane type* consists of a labyrinth formed with parallel metal sheets with suitable liquid collection "pockets." The gas, in passing between plates, is agitated and has to change direction a number of times. Obviously, some degree of centrifugation is introduced, for as the gas changes direction the heavier particles tend to be thrown to the outside and are caught in the pockets provided.

Coalescence of small particles into those large enough to settle by gravity is provided by two mechanisms: agitation and surface. The surface of the element is usually wet, and small particles striking it are absorbed. Inasmuch as the pockets are perpendicular to the gas flow, the liquid thus formed does not have to flow against the gas. Consequently, small compact units have a large capacity.

As the plates are placed closer together and more pockets are provided, greater agitation, centrifugal force, and collection surface are provided, but the pressure drop is increased correspondingly. Thus, for a given flow rate, the collection efficiency is normally some function of the pressure drop.

In the average application this pressure drop varies from 30-250 mm H₂O [1-10 in. H₂O] of water. Because of this pressure drop and to prevent gas bypassing the extractor, a liquid-collection pan incorporating a liquid seal is necessary for the liquid to drain properly.

When using the vane type mist extractor one must be careful that the pressure drop across it does not exceed its height above the liquid level, if a downcomer pipe is used. Otherwise, liquid will be "sucked out" overhead. The downcomer pipe can become partially plugged to accentuate the problem. Two wire mesh pads may be used in like fashion, with the first being used as a coalescer. As a rule-of-thumb, the coalescer pad should have about half the free space area of the second pad. Any wire mesh pad should be installed so that the flow is perpendicular to the pad face (pad is horizontal in a vertical vessel).

Space requirements for vane type mist extractors may be estimated by using the Equation 11.10 and sizing criteria below.

	K _s	
	Metric	English
Vertical	0.12 m/s	0.4 ft/sec
Horizontal	0.20 m/s	0.65 ft/sec

Increased use has been made of mist extractors composed of a *knitted wire mesh* supported on a lightweight support. This material has given generally favorable results and has a low installed cost.

The element consists of wire knitted into a pad having a number of unaligned, asymmetrical openings. Although similar in appearance to filter media, its action is somewhat different. The latter are rather dense and have small openings. This knitted wire, on the other hand, has about 97 to 98% free voids and collects the particles primarily by impingement.

The material is available in single wound units of varying thickness in diameters up to 90 cm, or in laminated strips for insertion through manholes in large process vessels.

The principle of separation is similar to that of the vane-type unit. The gas flowing through the pad is forced to change direction a number of times, although centrifugal action is not so pronounced. Impingement is the primary mechanism.

A liquid particle striking the metal surface, which it does not "wet," flows downward where adjacent wires provide some capillary space. At these points, liquid collects and continues to flow downward. Surface tension tends to hold these drops on the lower face of the pad until they are large enough for the downward force of gravity to exceed that of the upward gas velocity and surface tension.

Efficiency is a function of the number of targets presented. This may be accomplished by increasing the pad thickness, changing wire diameter, or the closeness of the weave.

The wire mesh normally used falls within the following range:

Wire diameter:	0.076-0.28 mm [0.003-0.011 in.]
Void volume:	92 to 99.4 percent
Density:	48-529 kg/m ³ [3-33 lb per cu ft]
Surface area:	164-1970 m ² /m ³ [50-600 sq ft per cu ft]

The most commonly used wire has a void volume of 97 to 98%, a bulk density of approximately 192 kg/m³ [12 lb per cu ft], a surface area of 328-410 m²/m³ [100 to 125 sq ft per cu ft], with a wire diameter of 0.28 mm [0.011 in.]. A pad thickness of 100-150 mm [4-6 in.] is sufficient for most separator applications, although thicknesses up to 900 mm [36 in.] have been reported. In separator service, 100-150 mm [4-6 in.] will normally suffice.

Any common metal may be used in these units, including carbon steel, stainless steel, aluminum, monel, etc. The pressure drop is a function of the entrainment load, the pad design, and gas velocity, but will not exceed 3 cm of water in the average installation. Because of this small pressure drop, the elements do not have to be "held down" and are normally only wired to the support grid to prevent shifting unless surging flow is anticipated.

Experience has shown that the support grid should contain at least 90% free area in order to eliminate any restrictions to liquid drainage. The pads are light in weight so that a light angle-iron support is adequate.

When both liquid and solids are present, a portion of the latter obviously will be scrubbed out. When only dry solids are present, the efficiency is substantially less. At the present time, though, this type unit is considered primarily for liquid removal.

Many vessel carryover problems are encountered. Foaming is a major culprit and requires more than simply better mist extraction. Most such problems develop by default. The vendor automatically uses his standard sizing curves and equipment, and the buyer assumes this will be good enough.

With glycols, amines, and similar materials, which tend to foam, we would normally specify a dual mist extractor – the lower one being of the vane type and the upper one being a wire mesh. A space of 150-300 mm [6-12 in.] would be left between them. The vane type will handle large volumes of liquid but is relatively inefficient on small droplets. It therefore serves as a bulk removal device (and helps coalesce foam). The wire mesh, which has limited liquid capacity, may therefore operate more effectively.

The capacity of a wire mesh unit is represented by Equation 11.10. A maximum value of $K_s = 0.107$ [0.35] generally is satisfactory.

With materials like glycol and amine, which wet metal very well, a Teflon-coated mesh may prove desirable. Remember, the liquid must be nonwetting in order to stay as droplets that can run down the wires and coalesce into bigger droplets. A wetting fluid will tend to "run up" the wires.

The above two types are part of a class of equipment called *impingement separators*. The most common types of these are shown in Figure 11.7.

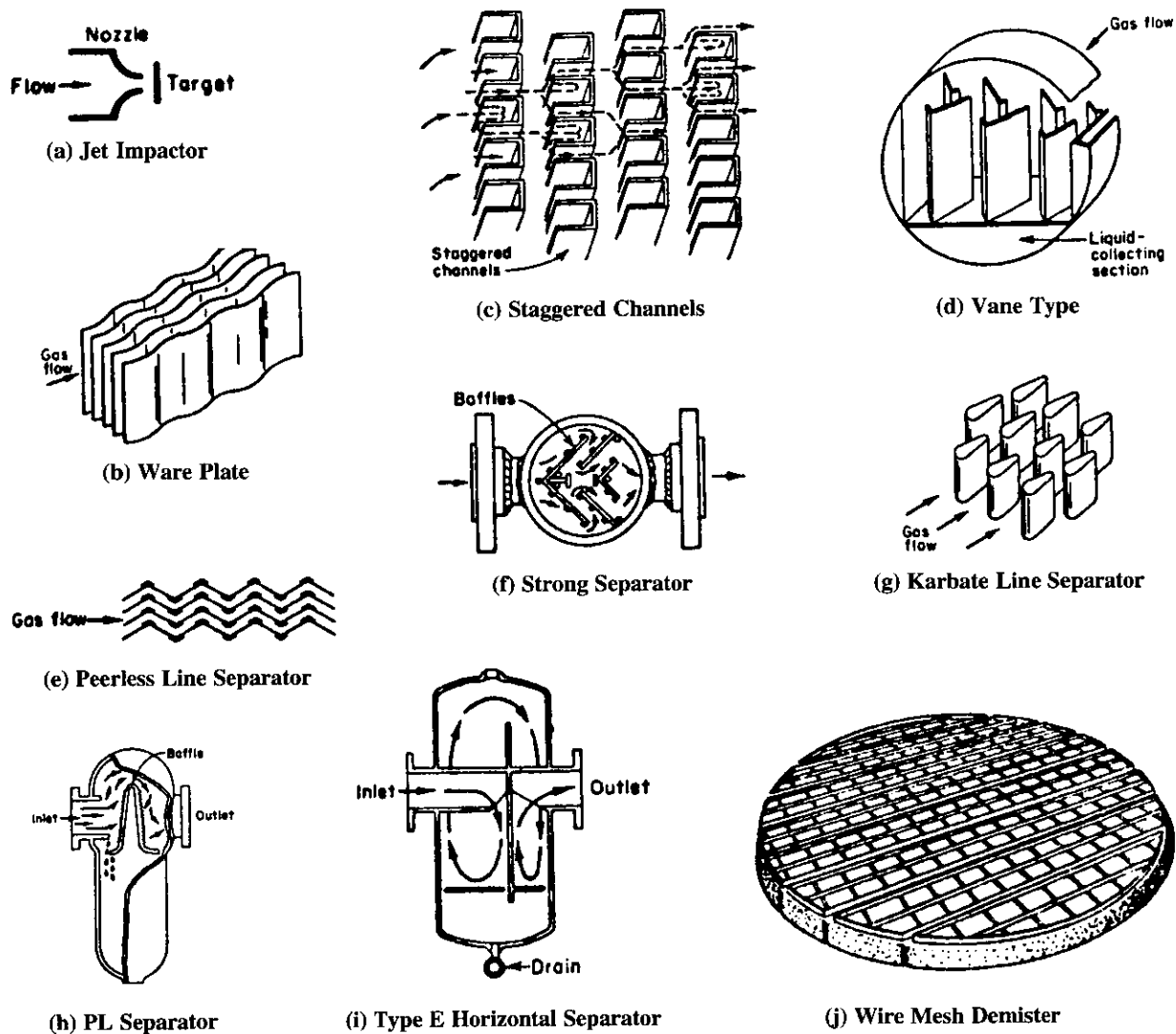


Figure 11.7 Examples of Typical Impingement Separators^(11.3)

Centrifugal Extractors (Separators)

It has been noted that centrifugal force is an integral part of separation processes. The standard oil and gas separator may have an inlet that utilizes centrifugal force to separate the larger droplets.

The same principle is used in some mist-extractor elements except that higher velocities are needed in order to separate the smaller droplets. The velocity needed for separation is a function of the particle diameter, particle and gas density, and the gas viscosity.

With a given system, the size of particle collected is inversely proportional to the square root of the velocity. Consequently, the success of a cyclonic mist extractor is dependent on the velocity attained. Furthermore, the velocity needed to separate a given size of particle must increase as the density of the particle becomes less. In addition to producing the necessary velocity, the mist extractor must provide an efficient means of collecting and removing the particles collected in order to prevent re-entrainment.

One common type of equipment is often called a "steam separator" since it has been widely used to separate condensate and pipe scale in steam systems. It is normally a relatively small vessel which imparts a high velocity to the incoming gas and then makes the gas change direction radically to prevent re-entrainment. In general, it will separate particles 40 μm and larger very efficiently.

Another type uses the same principle but, in addition, forces the gas to pass through a labyrinth which introduces impingement effects and forces the gas to change direction a number of times. This is, in reality, a combination type and is relatively efficient. The general performance characteristics are the same as efficient mist extractors of other types. Some, however, are complex and relatively expensive.

In the last several years the use of a multicyclone extractor has become more widespread. This may be mounted in the top of a regular vessel or in a small separate scrubber.

This unit imparts a centrifugal force to the entering gas. It spirals downward and reverses direction. Clean gas emerges while dust and distillates thrown to the wall by centrifugal force are discharged at the bottom.

By using a number of such tubes in parallel, a high velocity is imparted to the gas which would indicate that small particles could be separated. For a given flow rate, therefore, the number and size of tubes used depends on the relative density of the particles and the gas.

Separation becomes more difficult as the gas density increases. This probably accounts for the lower efficiency exhibited by all centrifugal devices at elevated pressures. However, all other devices have this same failing. This common fault points out that increasing pressures give a corresponding increase in the extraction problem.

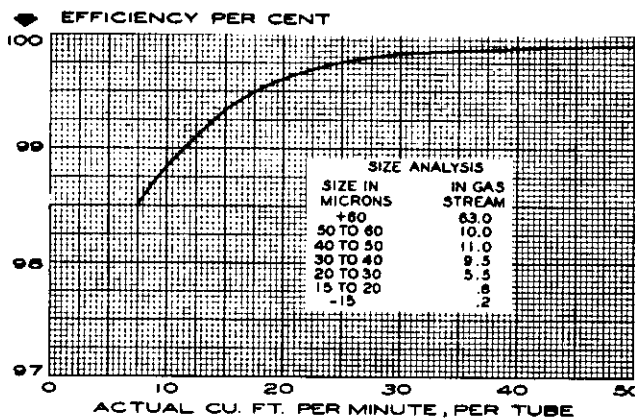


Figure 11.8 Effective Efficiency of a Centrifugal Mist Extractor

Figure 11.8 shows the overall collection efficiency when processing pipeline dust. The recommended flow rate for such devices is usually proportional to ΔP to the 0.5 power. So, efficiency is obtained at the expense of pressure drop.

The multicyclone separator has been used efficiently in the separation of dusts as well as liquid particles. It is substantially self-cleaning and can handle relatively large quantities of both. In some cases an impingement-type element has been effectively used ahead of such a unit. The first acts as an agglomeration agent for the smaller particles which may then be separated in the centrifugal unit.

The chief disadvantage of any centrifugal unit is that the efficiency falls off rapidly as the velocity decreases below a certain point. Therefore, when widely varying loads are encountered, this type of unit shows decreased advantage. However, if a higher pressure drop is tolerable the unit may be designed for the lowest flow rate anticipated since it gives efficient separation at greater than rated capacity.

Figure 11.9 shows some of the typical centrifugal separators discussed above as well as some using a combination of impingement and centrifugal force.

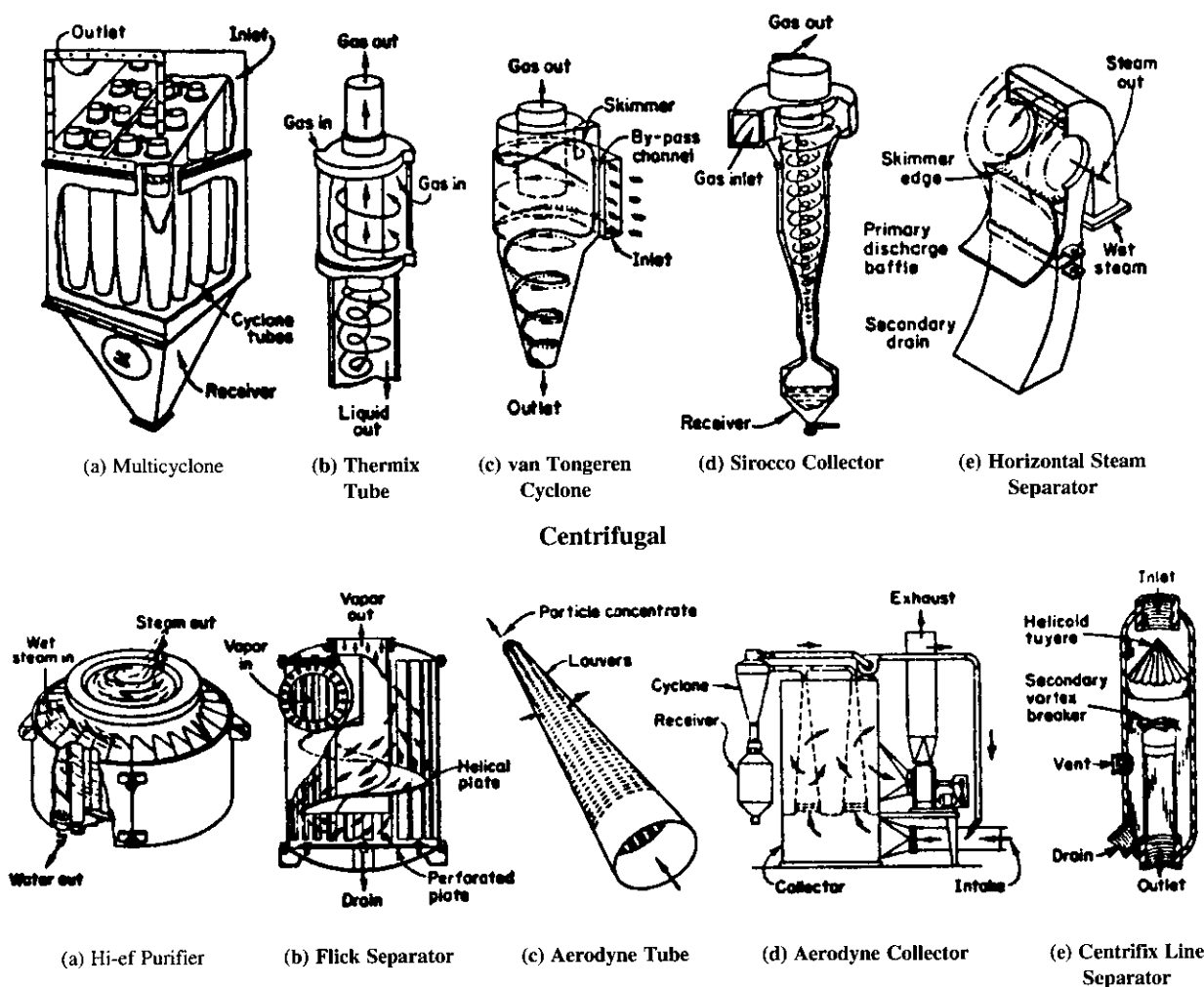
Centrifugal and Impingement^(11.3)

Figure 11.9 Examples of Centrifugal and Combination Centrifugal-Impingement Separators

Nozzles

Nozzles must be of a size to minimize erosion/corrosion, pressure drop, entrainment, etc. The equations following may be used to estimate nozzle sizes.

$$v_i < \frac{A}{(\rho_m)^{0.5}} \quad (11.21)$$

$$v_g < \frac{B}{(\rho_g)^{0.5}} \quad (11.22)$$

$$v_L < C \quad (11.23)$$

Where:

v_i, v_g, v_L

ρ_m, ρ_g

A

B

C

Metric	English
m/s	ft/s
kg/m ³	lbm/ft ³
60	50
75	60
1.0	3.3

Liquid outlets should be equipped with anti-vortex devices to prevent gas from going out with the liquid. Several types are shown in Figure 11.10.

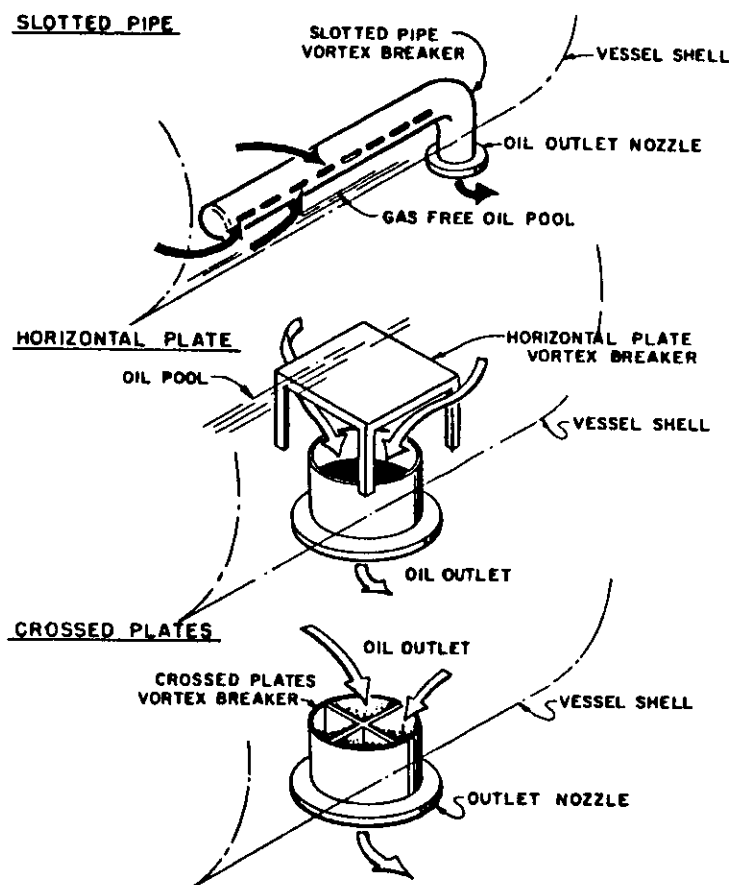


Figure 11.10 Several Types of Anti-Vortex Devices

There is a tendency for vendors to use too small, and maybe too few, liquid outlets. On very large (and long) horizontal separators two liquid outlets may be desirable, particularly if surging is present.

EMULSION TREATERS, CLARIFIERS AND COALESCERS

These are all devices that accelerate the separation of two phases when the natural retention time is too long for commercial application.

Emulsion treaters use some combination of heat, electricity, chemicals, separation time and coalescence to separate oil and water. The purpose may be to reduce the *B.S. & W.* (Basic Sediment and Water) content to meet specifications. Or, the purpose may be to meet salt content specifications. These are related because most of the salt is in the water present. Most U.S. refineries have desalting equipment. The majority of non-U.S. refineries specify a maximum amount of salt of 10 lbm per 1000 API barrels of oil.

In emulsion treating there are two basic strategies. One is to use heat and/or chemical to reduce the interfacial tension in the outer film of the small droplets in the emulsion (usually water in oil). This allows the drops to coalesce into a size large enough to separate by gravity. A typical U.S. treater uses a gas fired indirect heater unit. Chemical may or may not be necessary. One or more "hay" (coalescing) sections may be used.

The second strategy uses an electrical field in what are called *electrostatic heaters*. These often are called *desalters*. They essentially are two vessels in one. The first section is like an ordinary oil treater which uses heat and/or chemical to break some of the emulsion. The effluent from this section flows to the electrode section. The electrical field causes rapid collision between drops of water which coalesce into a size capable of settling by gravity. Some desalters contain what amounts to two electrode sections. In the second, 3-10% relatively fresh water is added, hopefully to dilute salt in the entrained water.

The basic desalting strategy depends on whether or not low salinity water is available for dilution. Detailed mechanical design is critical. The design of the injector and mixing valve must be determined by experience and can have a significant effect on the efficiency of desalting.

Separation of two liquid phases is often conducted at low pressure so it is economical to have large diameter vessels. For separation of oil and water, a *wash tank* may be used. This utilizes mere gravity separation, with or without the addition of fresh water.

Low pressure vessels are also effective for the separation of oil-glycol solutions. Separation is slower in a sour system than in a sweet system because iron sulfide particles float on the interface and hinder movement between phases.

An empirical equation for estimating retention time for *wash tanks* and general liquid-liquid vessels is:

$$T = A \left[\frac{\mu}{(\gamma_b - \gamma_t)} \right] \quad (11.24)$$

Where: T = retention time, hours
 μ = viscosity of continuous (predominant) phase, cp
 γ_b = sp gr of bottom phase (water solution)
 γ_t = sp gr of top phase (oil)
 A = constant which varies from 0.05-1.0 in most service, determined by test

Equation 11.24 *does not* apply when artificial coalescing is accomplished by electrical force, contact surface, centrifugal force, heat and chemical, etc.

Retention times are seldom less than 8 hours, even for fairly light crudes, and can exceed 24-36 hours for heavy crude oils ($^{\circ}\text{API} < 15^{\circ}$)

Clarifiers are used primarily to separate small amounts of oil from large amounts of water by air flotation, flocculation, or similar approach. A *coalescer* is any element or vessel whose primary purpose is to provide a large amount of surface for the agglomeration of small droplets. A mist extractor is one form of a coalescer.

GAS CLEANING

Cleaning normally refers to the removal of small solid particles and liquid entrainment from a gas stream. There are six basic types of gas cleaning equipment – coalescer separators, dust filters, gas-liquid separators, centrifugal scrubbers, electrostatic precipitators, and oil-bath scrubbers. All are, in effect, two-section devices – bulk removal followed by some method of removing the "fines."

Dust Filters. No discussion of mist and dust removal is complete without at least some mention of filtration. This is most often a pipeline application but the problem originates in the production system.

A filter differs from the mesh-type mist extractor in that the element is composed of rather densely packed woven elements that trap particles on the face of the pad. The close clearance between the fibers allows the gas to pass while the larger particles build up on the surface and to some degree in the bed of the

filter itself. A number of materials have been used, including glass fiber, Dynel, felt, and similar materials that may be readily woven, or compacted, and formed into pads.

The primary use thus far has been for the elimination of dust in lines. One of the earliest applications was a "bag filter" downstream from dry-desiccant dehydration plants which, as the name implies, was primarily a porous woven bag held in place in the line. This rather crude attempt at dust separation was often a nuisance. As a cake of dust built up, the pressure drop became excessive and the bags often failed. Furthermore, the line had to be shut down periodically to permit cleaning of these filters.

Felt, compressed into pads, has also been used. The most successful applications had several layers of felt pads held in compression, with the pads being placed parallel to the flow. All of these materials, however, were difficult to clean and tended to deteriorate under the action of the gas, particularly in the presence of liquids. Part of this disadvantage has been overcome by the use of synthetic materials, such as glass fiber, which are more impervious to organic liquids.

Even with these materials, though, liquid separation appears low even though dust may be efficiently separated. Liquid particles appear to agglomerate on the downstream side of the pads as a solid film of liquid. As the gas passes through this film, bubbles are formed, burst, and are re-entrained. At this point the problem is the same as the original one except that the dust is gone, and the liquid particles are larger and may be easier to catch in a conventional mist extractor. Observations such as this have led to the conclusion that the best system might well consist of a scrubber with both filter and mist extractor elements.

A filter element ideally should have the following characteristics:

1. Be self-cleaning to minimize down time
2. Be readily removable for cleaning
3. Be impervious to the action of organic liquids and water so that it does not swell or break down
4. Have structural strength and be rigid enough to hold its shape
5. Have a relatively low pressure drop
6. Have a surface that is "nonwetted" by the liquids encountered to prevent the liquid from "crawling" through the element
7. Be so mounted that scale, drilling mud, and other large contaminants encountered are separated prior to the gas passing through the pad
8. Be able to separate dust and liquid simultaneously with high efficiency

Needless to say, the above animal is not in captivity. Many filters used have worked excellently in one location and given poor results in another. The results obtained with a given field installation are often difficult, if not impossible, to evaluate. After installation of a given filter a previous problem sometimes disappears, but one is never certain whether the filter or one or more other changes in the system is responsible. This undoubtedly accounts, in part, for the erratic results obtained.

One type of commercial filter available on the market is composed of multiple circular elements, the number depending on the area needed. The gas flows into the top of the filter, through the elements, and out through the tubes. Dust is retained in the element and liquid coalesces into larger particles for removal in a mist extractor that may be located downstream. The end flange is removed for replacement of the elements. This type of filter offers the advantage of a high surface area per unit volume of vessel. The amount of filter area needed is a function of the material, its density, the gas specific gravity, and the method of construction.

A *filter separator* of the type shown in Figure 11.11 contains an inlet section with molded fiber glass elements which are hollow in the middle. The liquid coalesces in these elements and flows to the center section. The outlet section is, in effect, a large mist extractor. Manufacturers sometimes claim almost 100% removal of particles down to 1 micron. This is technically feasible, but may not be achieved in practice.

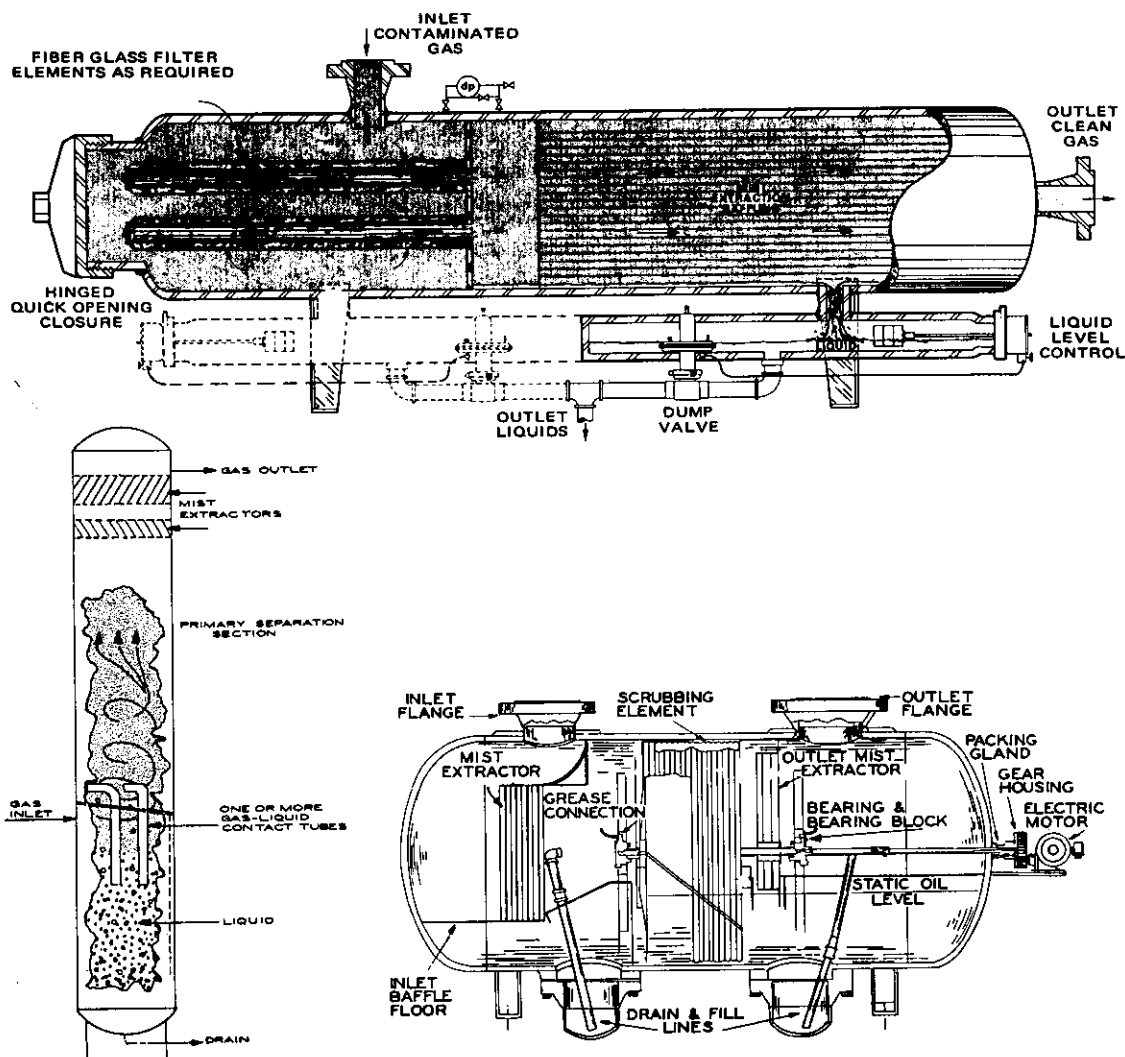


Figure 11.11 Some Typical Filter Separator-Scrubbers

Sizing is critical. Remember that the "back-end" of a filter separator is a vane type mist extractor and the diameter should be based on sizing criteria presented earlier for this device. Some vendors propose smaller vessels based on the efficiency of filter separation. It does no good to coalesce small droplets if they are re-entrained in the mist extractor.

The very traits which enable the filter separator to remove small sizes also enable it to be plugged by "dirty" streams. If the filter plugs it can fail. The application in gas processing primarily is for:

1. Removal of amine and glycol carryover
2. Removal of hydrocarbons ahead of amine, glycol and solid desiccant units
3. Removal of desiccant dust
4. Removal of solids and liquids ahead of heat exchangers and compressors to eliminate fouling, and reduce maintenance

Liquid Scrubbers

All liquid scrubbers are simply vessels designed to provide intimate contact between the gas to be cleaned and the scrubbing liquid. The general design shown in Figure 11.11 is typical of the vertical type.

Except for the lower section, the general features are similar to those of oil and gas separators, a primary separation and mist extraction section being provided. Many different internal arrangements have been used in the scrubbing section, but the most common has the gas flow through the liquid. Their contact is aided by some sort of contact device. This liquid is then removed in the separation section and returned by gravity to the liquid sump.

Other arrangements are used, but they usually involve the use of an external power source. Figure 11.11 also shows a horizontal vessel divided into two sections. The first acts primarily as a liquid knock-out, while the other is the liquid scrubber. The latter consists of corrugated cylinders which are rotated by an external motor.

The lower portion of the vessel is filled with oil, and as the cylinders rotate they become coated with oil. Dust passing through the oil labyrinth formed by the cylinders will tend to cling and is subsequently removed when the rotating cylinder passes through the oil. As expected, this unit is reported to have about three times the capacity of an equivalent vertical unit, but has the obvious disadvantage of requiring an external source of power.

A properly designed liquid scrubber is relatively efficient, but has the disadvantage of introducing liquid downstream. Carryover is normal but sometimes entire charges of scrubber liquid have been lost. This carryover leads to the danger of hydrostatic lock in compressor cylinders and damage to metallic packing. Where fiber compressor valve plates are used there is often danger of softening, causing early failure. The normal carryover is, of course, limited by the efficiency of the mist extractor used.

Abnormal loss is usually due to one or more of the following:

1. Dilution of the scrubber oil by incoming liquid or absorption of components from the gas
2. Foaming
3. Surges of liquid due to rapidly fluctuating gas loads and pressures
4. A combination of gas flow rate and pressure that exceeds the rated capacity of the vessel

The first problem may be minimized by putting a conventional "dry" scrubber ahead of the liquid scrubber, but this introduces obvious economic drawbacks. Some have used a low-vapor pressure liquid immiscible with oil such as diethylene glycol, periodically or automatically skimming off any oil collected. There are some successful installations of this type, but glycol is relatively expensive. Except for very rich gas, the absorption of light components is negligible.

Foaming may result for various reasons, among which are the presence of aromatics or foreign liquids such as those injected into wells and pipelines to minimize corrosion.

Liquid surging may be largely prevented by proper design and operation of the line. Provision for bypassing the scrubber during blowdowns, etc. is recommended.

Line pressures normally vary, due partially to the widespread practice of packing lines during periods of off-peak demand. Consequently, such vessels should be sized at the minimum and maximum pressure and maximum flow rate anticipated. One reported investigation disclosed that the abnormal losses of oil were obtained at operating pressures 12% over design with the flow rate 16.5% above capacity. However, at an operating pressure of 37.7% over design and a throughput only 2.5% over design capacity, the oil losses increased about 500%. It is because of these carryover problems that many operators have resorted to dry-type scrubbers, although there are many successful liquid units in operation.

Scrubber Liquid. The liquid used should have a low vapor pressure, low pour point, relatively low viscosity, and be able to "wet" the dust. One manufacturer recommends any oil having the following general properties:

Initial boiling point:	260°C [500°F] minimum
Endpoint:	427°C [800°F] maximum
Gravity:	25°-35°API
Viscosity:	100 SSU at 40°C on lean gas below 3.5 MPa 150 SSU at 40°C on rich gas and/or above 3.5 MPa

Most dust and debris in pipelines is efficiently scrubbed with most any liquid, but two cases are known where material from the producing formation went through two separators and a glycol dehydrator containing four contact trays and then accumulated in a desulfurizer absorber. Fortunately, this is the exception rather than the rule.

Equation 11.11 may be used for sizing liquid scrubbers. A K_s value of 0.16 is recommended for pressures around 6.9 MPa [1000 psia] with vertical scrubbers; about 0.45 is suitable for horizontal scrubbers.

The last term of Equation 11.10 indicates that the separation of liquid and vapor becomes more difficult as their densities become closer. The density of the liquid is substantially independent of pressure, while that of the vapor is roughly proportional to pressure. Therefore, as the pressure is increased, the last term becomes smaller. However, this is more than compensated for by the increase in P . Consequently, Equation 11.11 would indicate an increase in capacity with pressure.

This is contrary to the results sometimes found in actual practice. In fact, the liquid carryover problem has generally become more pronounced as scrubber pressures have increased. Therefore, it would appear that the coefficient in Equation 11.10 also varies with pressure. The value used has been found satisfactory at pressures to 6.9 MPa [1000 psi], however. For other than estimation purposes, reference should be made to guarantee capacity curves of the manufacturer.

Regardless of capacity, it takes clean oil to produce clean gas. Therefore, an external source of oil is recommended where economically feasible. As the oil loads with dirt a type of vapor-liquid-solid equilibrium occurs. Dirt also promotes stable foaming.

CONTROLS

Control of pressure and level are basic to good separator operation. The choice of control modes, sensitivity characteristics, control hardware, etc. depends on the purpose of the separator and the process modules immediately preceding and following it.

As a general rule, the pressure of the separator must be held rather constant, independent of the operation of adjacent equipment. Usually this means a back-pressure valve on the gas outlet. The proportional control mode (without reset) normally will be satisfactory. Any offset due to flow rate changes normally presents no problem. However, the vessel design pressure and high pressure alarm or shut-down controls must be consistent with the range of pressure expected for the proportional setting and offset anticipated.

A deviation of 10% from the pressure set point can occur, although 6-7% is a good design number. If a high pressure alarm is intermediate between normal operating pressure and high pressure shut-down, there must be a broad enough pressure range allowed for operator interaction to correct the problem.

The residue gas often is suction to a booster compressor. If this is a reciprocating compressor or an axial compressor, a back pressure valve definitely is needed. No valve is absolutely required with a centrifugal compressor, since the suction pressure oftentimes can be kept rather constant with compressor controls.

Check anticipated conditions carefully, particularly variations in gas flow rate and compressor discharge conditions. If you leave out the valve you may wish to add a "spool" in the line so one could be added later.

If efficient vapor-liquid separation is the primary purpose of the vessel, the liquid level should be held relatively constant. As per the principles outlined in Chapter 9, proportional plus reset control may be necessary. The problem is to allow continuous throttling of liquid while accommodating liquid surges that may occur.

Figure 11.12 shows a split-range approach often used. So long as liquid input is rather steady only Valve A is operating. It is fully closed at pilot output pressure of 3 psi and fully open at 9 psi. It would be set at a low percentage of proportional control to minimize offset.

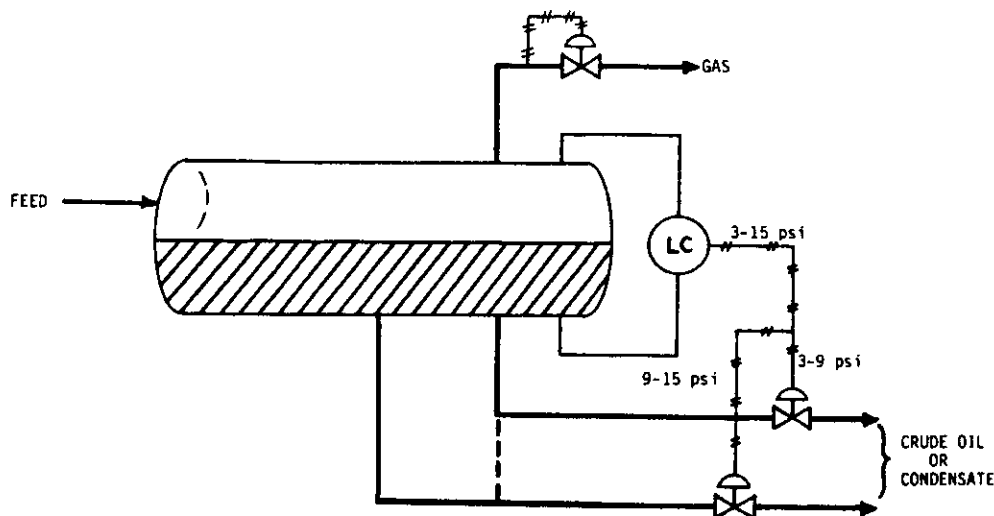


Figure 11.12 Example of Split-Range Level Control

A large slug of liquid hits the separator. The level rises, pilot output pressure rises toward 9 psi, Valve A is fully open, but the level continues to rise. At 9 psi, Valve B starts to open to relieve the surge. How fast it opens depends on the setting (% proportional or snap action). Once the crisis is over and the level is back within normal range, Valve B closes and waits for the next such upset.

This system provides sensitive routine level control plus the added capability for relieving surges not possible with a single valve system. Notice that the line to Valve B may come off the same liquid nozzle as Valve A or from a separate one at another point in the vessel. In a horizontal separator above 20 ft in length, two liquid nozzles may be a justifiable expense. The one closest to the inlet can alleviate a surge wave before it reaches the level controller and provide stable control. In a vertical separator one would always use a single liquid nozzle.

Check the size of the liquid nozzles. They tend to be too small (for competitive cost reasons) even though they may technically meet specifications like those of the API for steady-state flow. Inadequately sized nozzles for surging flow is a surprisingly common "control" problem.

Suppose that the liquid is being pumped out the bottom of the separator. Figure 11.13 shows three different arrangements. In (a) the level control valve simply changes the pump back-pressure to match pump output to level set point. A relief by-pass circuit is provided to protect the pump from over-pressure by the valve closing too much. A second alternative (b) is to provide a rate controlled by-pass valve. This is more expensive than (a) because of the extra valves and controls needed.

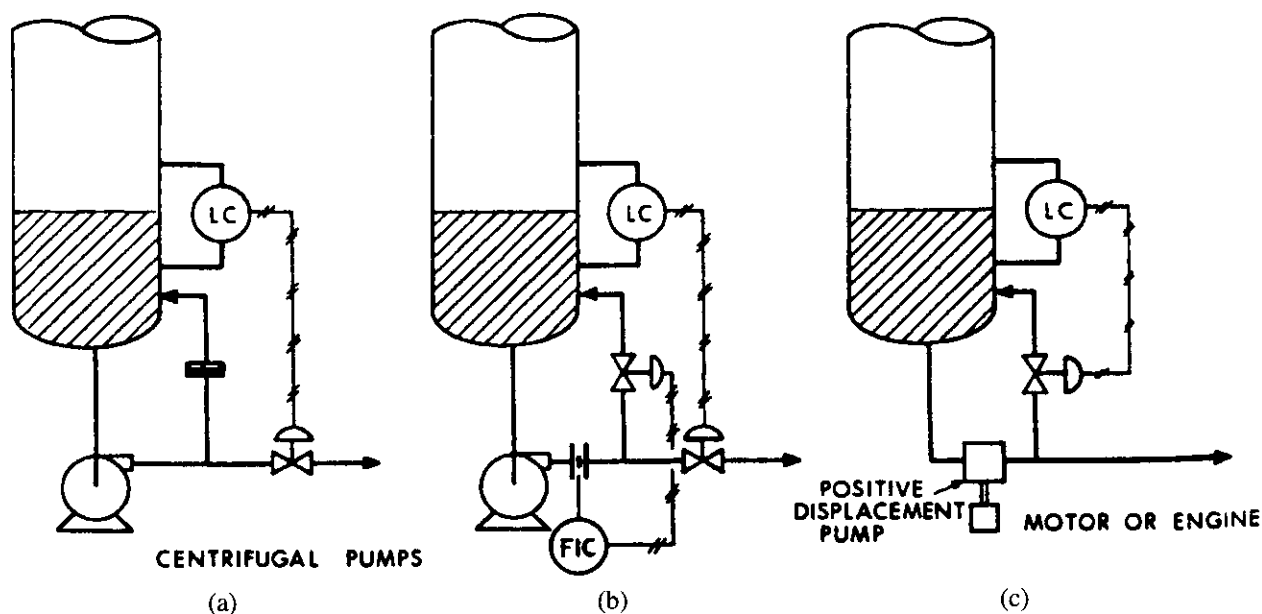


Figure 11.13 Example of Level Control with a Pump

Both (a) and (a) are inefficient from a power utilization standpoint. Throttling and bypassing utilize power that serves no useful purpose. In some installations it is desirable to use a variable speed drive. The level controller then adjusts speed to maintain level.

As shown in (c), it is not possible to throttle a positive displacement pump. The level will control by-pass rate or pump speed.

Liquid Residence Time and Controls

Final separator design must conform to residence time requirements that are compatible with the control system. Figure 11.14 shows a horizontal separator equipped with high and low level shut-down (LSDH and LSDL) as well as high and low level alarms (LAH and LAL). These are shown in relation to the normal liquid level (NLL).

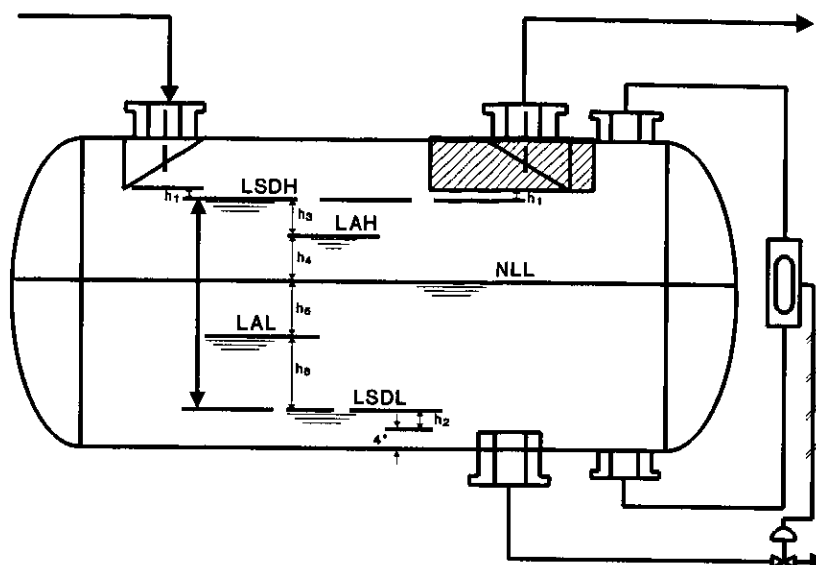
If surging flow is expected, enough volume must be provided on both the low and high side so that the alarms do not trigger routinely. Once an alarm has been activated, enough time must be provided for the operator to take corrective action before shutdown occurs.

The bottom portion of Figure 11.14 provides a rule-of-thumb guideline for the residence time between levels. The numbers in each square are in minutes.

It should be apparent that separator and control design should be compatible. A substantial portion of separator volume may be required for control purposes. Failure to properly coordinate process and control functions results in too frequent alarms and shut-downs and/or inefficient separation.

Three-Phase Separation Control

Figure 11.15 shows various ways of controlling the high pressure separation of gas, oil and water. Shown are the use of interface controls, buckets and weirs. In *interface control* a displacement float reacts to the different density of the two adjacent liquid phases. Only this difference affects buoyancy change with level. Thus, it may be very unstable. *Buckets* are chambers within the vessel where one or more liquid phases are segregated. *Weirs* are used to aid in segregation to eliminate the need for interface controls.



Time Between Control Points (min)					
	LSDL	LAL	NLL	LAH	LSDH
LSDL	X	6	8	—	—
LAL	5	X	2*	—	—
NLL	6.5	1.5*	X	2*	8
LAH	—	—	1.5*	X	6
LSDH	—	—	6.5	5	X

* Time and volume between LAL and LAH may be set by surge volume to be accepted by vessel.

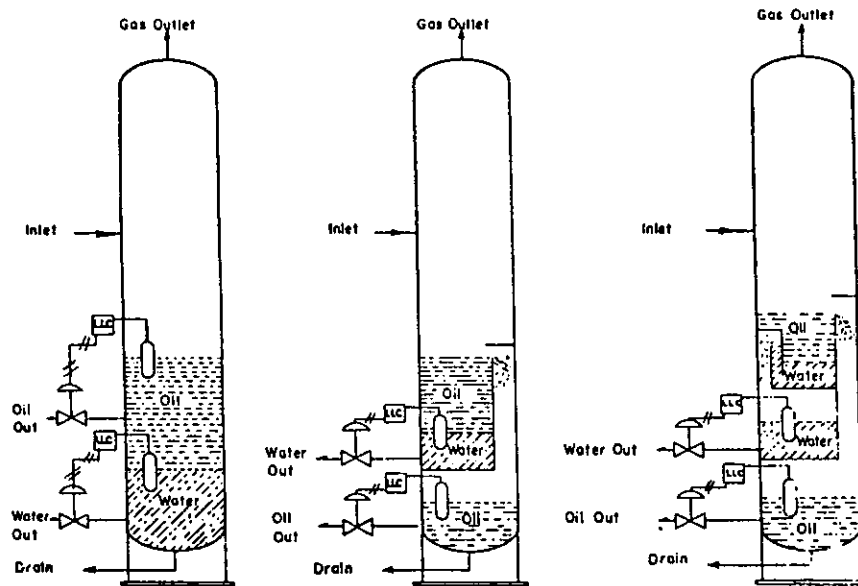
Values to the right of diagonal are for surging flow or feed to a fired heater.
Values to the left are for normal operation.

Figure 11.14 Guidelines for Residence Time Provision

Methods (3) and (4) of Figure 11.15 are recommended even though they have the highest initial cost. All control interfaces are gas-liquid to provide maximum control stability.

A similar control system applies for treating vessels and free water knockouts.

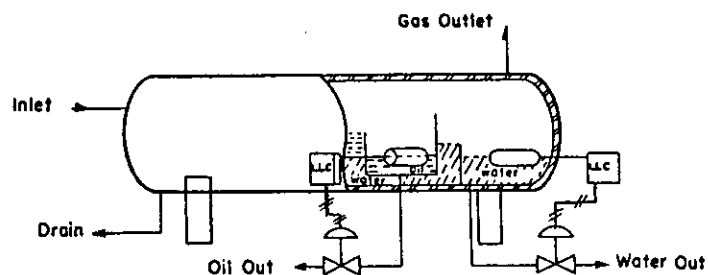
Another type of three phase separator design involves a boot-type separator as shown in Figure 11.16. These separators are frequently used in glycol hydrocarbon separation service in LTX and refrigeration type



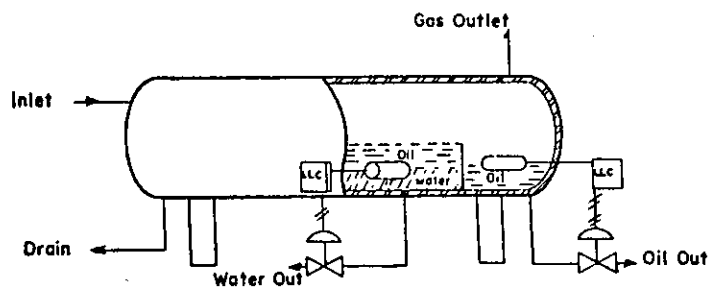
Method 1: Separation obtained through level control only.

Method 2 Separation obtained by interfacial level control and weir

Method 3 Weir used to separate oil and water to avoid use of interfacial controls.



Method 4 Operates in same manner as vessel in Method 3.



Method 5 Operates in same manner as vessel in Method 2.

Figure 11.15 Various Methods for Controlling Three-Phase Separation

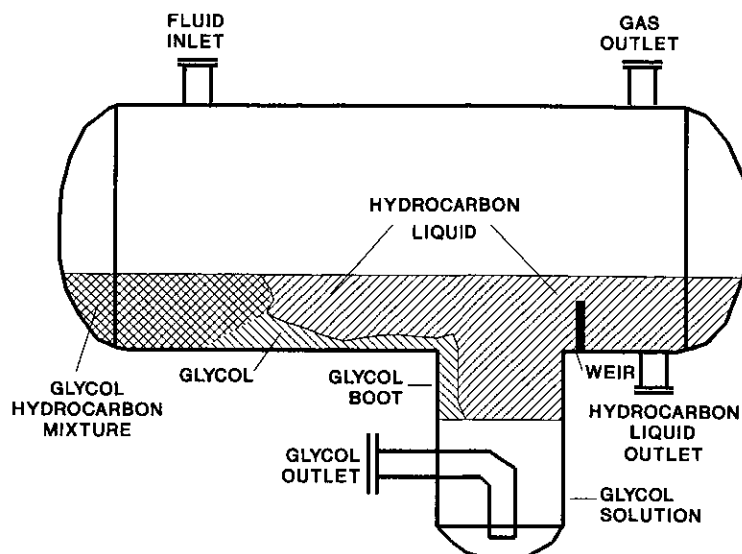
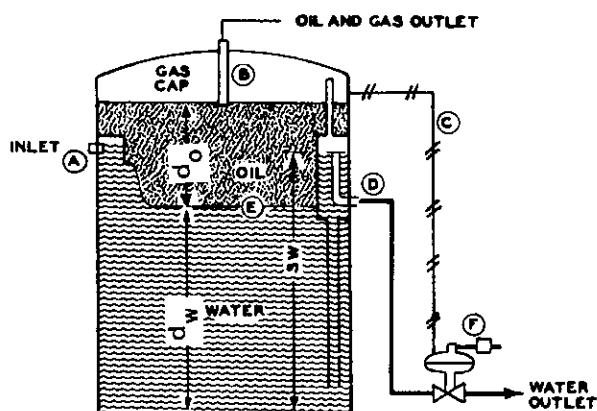


Figure 11.16 Horizontal Three-Phase Separator

plants. The level of the glycol phase is controlled with an interphase controller, while the hydrocarbon level is controlled with a normal level controller. These separators are also used in oil/water service.

In the low-pressure, free-water knockout, the free-water settles out to the bottom and the oil and gas pass overhead – usually to a treater. The outlet B is extended into the vessel to insure the presence at all times of a gas cap and to control the oil level.



The location of the oil-water interface E is fixed by the position of adjustable nozzle D. Since the water outlet box is vented into the gas cap, rotation of this nozzle changes the interface level for:

$$d_o \rho_o + d_w \rho_w = S_w \rho_w + \Delta P_f \quad (11.25)$$

Where: d_o = thickness of oil in vessel
 d_w = height of water in vessel proper
 S_w = height of water in outlet box
 ρ_o, ρ_w = density of oil and water respectively
 ΔP_f = pressure drop in downcomer pipe and outlet box

In a well designed system operating at normal capacity, ΔP_f may be neglected. Equation 11.25 may then be re-arranged to read:

$$d_w = S_w - d_o \left(\frac{\rho_o}{\rho_w} \right) \quad (11.26)$$

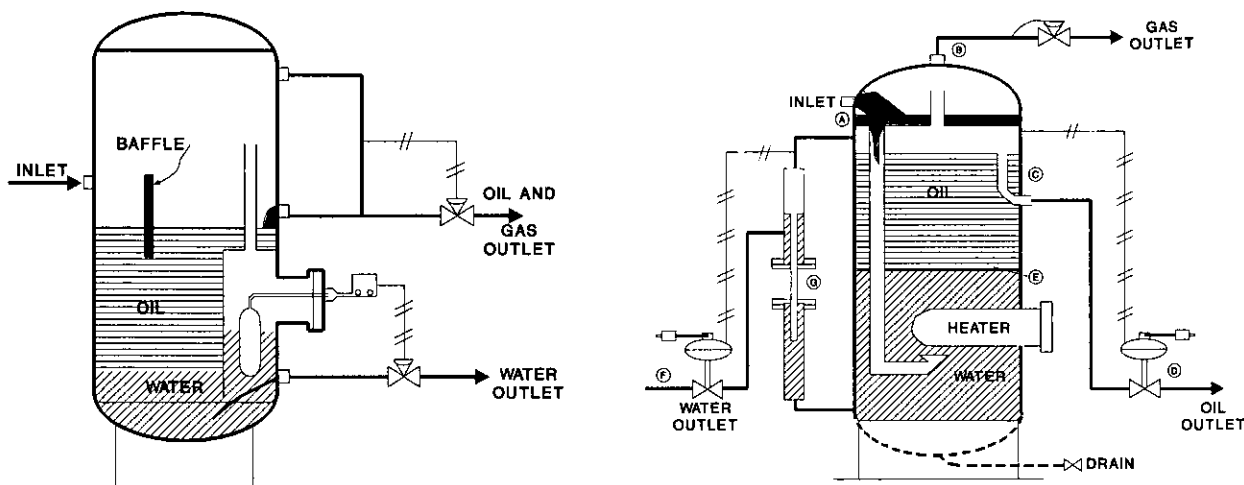
Consequently, any change in S_w will cause a corresponding change in d_w and d_o . As S_w is decreased, d_o gets larger, so that small changes in S_w give correspondingly larger changes in d_w .

Water is removed from the vessel with a simple weight-loaded, self-acting diaphragm valve. With the top of the diaphragm vented into the vessel gas cap, the only force acting on the underside of the diaphragm

is the head of liquid. The weight is then simply adjusted on the arm to compensate for the desired liquid head operating the valve.

This is really a form of back-pressure service, but only a valve designed specifically for such service should be used.

Shown at left below is a similar system for a treater. In this case, though, an additional valve is necessary to remove the oil which is separated from the gas in the top section. This valve operates in the same manner as the water valve.



The water syphon G is a type commonly used and the principle is the same as the rotating nozzle. It also must be vented back into the gas space. The oil outlet is a fixed opening so located that a gas cap may be preserved below the top plate.

The control system shown with the high pressure free-water knockout to the right on the next page is similar to that for the low pressure unit. In this case, though, a conventional liquid-level controller is incorporated into the water-outlet compartment.

The oil simply overflows into the gas-outlet line with the combined stream passing through the gas back-pressure valve. Because of liquid surging, the controlling connection should be upstream from where streams commingle.

SUMMARY

The foregoing should make it clear that choice of separation vessels is never routine. As a general rule, the quality and reliability of your production/processing system will be determined to a large degree by the quality of your separation decisions.

REFERENCES

- 11.1 Barker, W. F., *Oil Gas J.* (Dec. 27, 1982), p. 186.
- 11.2 Watkins, R. H., *Hydr. Proc.* (Nov. 1967), p. 255.
- 11.3 Perry, J. H., et al., *Chemical Engrs. Handbook*, 4th Ed., McGraw-Hill, New York (1963).
- 11.4 Maher, J. L. and Coggins, R. W., *World Oil* (Aug. 1969), p. 29.
- 11.5 McMinn, R. E., *Ibid.* (Oct. 1969), p. 100.
- 11.6 Philbeck, W. T., *Ibid.* (Nov. 1969), p. 98.

NOTES:

APPENDIX 11A

This appendix contains some general correlations from the references shown below for estimating weight, size and relative cost of various equipment. Costs are a function of time and size. Indexes are published regularly showing price changes with time. The *Oil and Gas Journal* publishes one of particular use in the petroleum industry.

The change of cost with size or capacity quite often is a straight line on a log-log plot within a given size range which can be represented by the equation

$$C_1 = C_2 \left(\frac{x_1}{x_2} \right)^a$$

Where: C_1 and C_2 = the cost of two different units
 x_1 and x_2 = capacity (size) of these units

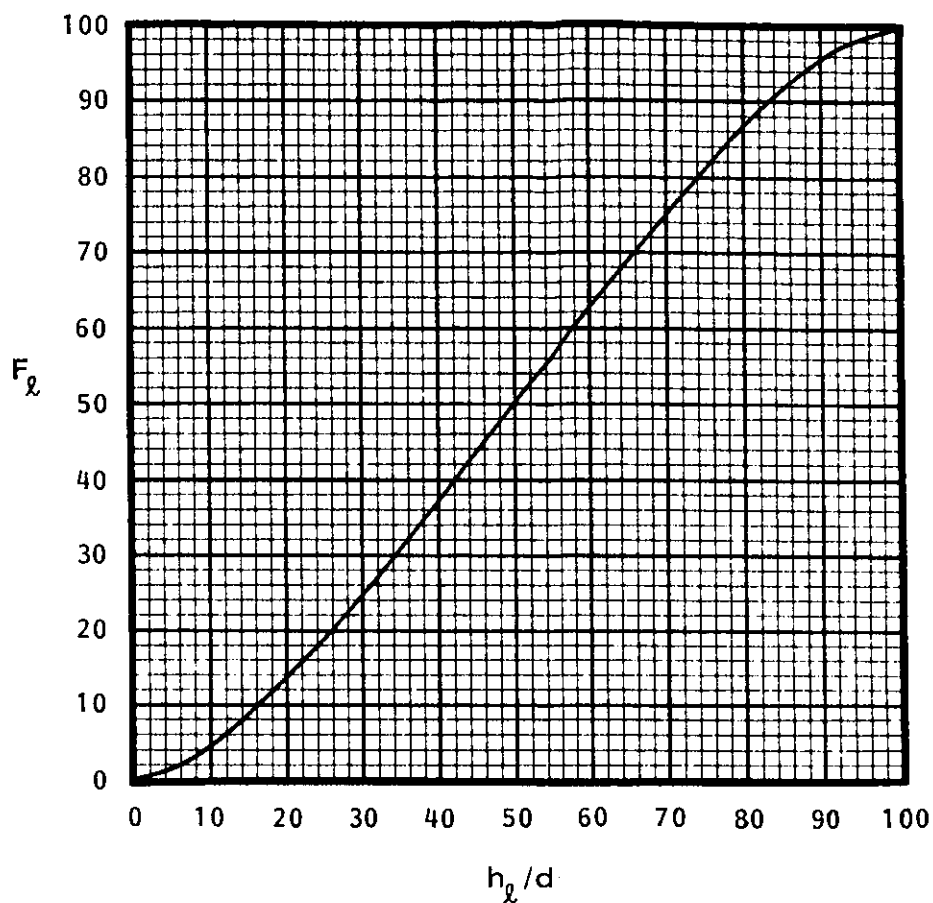
Typical values of "a" are shown below.

Equipment	Value - "a"	Limits of Applicability
Steel pressure tanks, 15-100 psia	0.54	1-100 x 10 ⁶ U.S. gallons
Liquid pumps, centrifugal	0.15	0.1-10 hp
Liquid pumps, centrifugal	0.60	10-1000 hp
Liquid pumps, triplex reciprocating	0.60	1-1000 hp
Fractionation towers	1.00	10-100 in. diam. vs. price/ft
Low pressure storage tanks (steel)	0.54	1-100 x 10 ⁶ U.S. gallons
Heat exchangers	0.59	10-20 000 sq ft surface
Fans, blowers, compressors	0.66	10-10 000 hp
Refrigeration units	0.66	10-500 tons rating
Liquid filters	0.56	10-1000 sq ft filtering area
Furnaces, tubular	0.54	0.1-100 MM Btu/hr duty
Furnaces, Dowtherm	0.35	0.1-3.0 MM Btu/hr duty

The tables and figures which follow are useful approximations for planning purposes only. Please note that some of the weights are dry weights and some include normal liquid weight during operation.

References:

1. Lilly, L. L., private communication.
2. Mayer, J. L. and Coggins, R. W., *World Oil* (Aug. 1969), p. 29.
3. McMinn, R. E., *Ibid.* (Oct. 1969), p. 100.
4. Philbeck, W. T., *Ibid.* (Nov. 1969), p. 98.

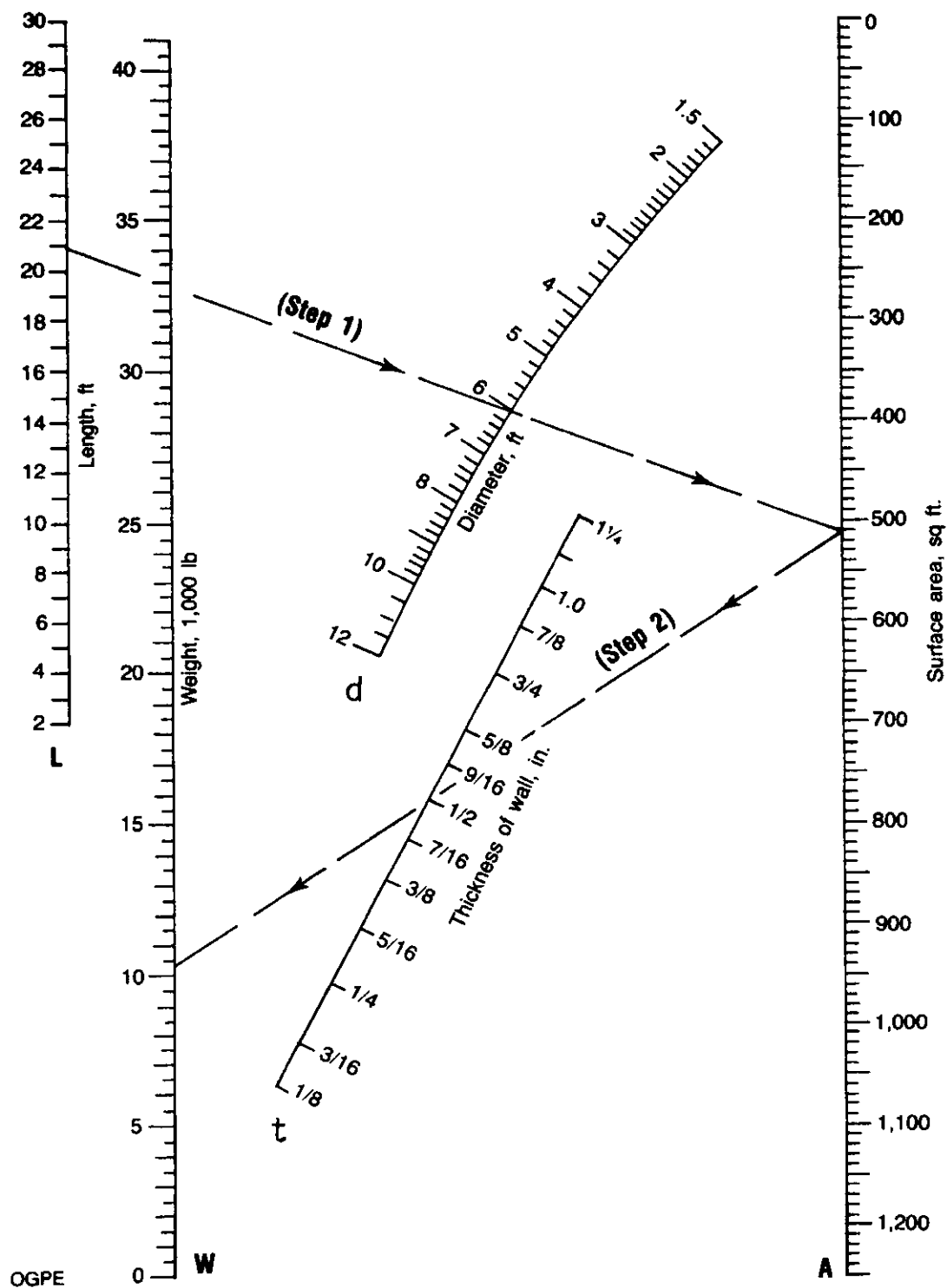


The above figure is for a horizontal, circular vessel. F_l (on the ordinate) is the area occupied by liquid as a percentage of total cross-sectional area. h_l is the total depth of liquid as a percentage of the total diameter.

The "F" for use in separator equations for fraction of area occupied by gas is

$$\frac{100 - F_l}{100}$$

For three-phase separation with two liquid phases, the F_l for the lightest liquid phase can be found by subtracting F_l for the heaviest phase from total F_l .



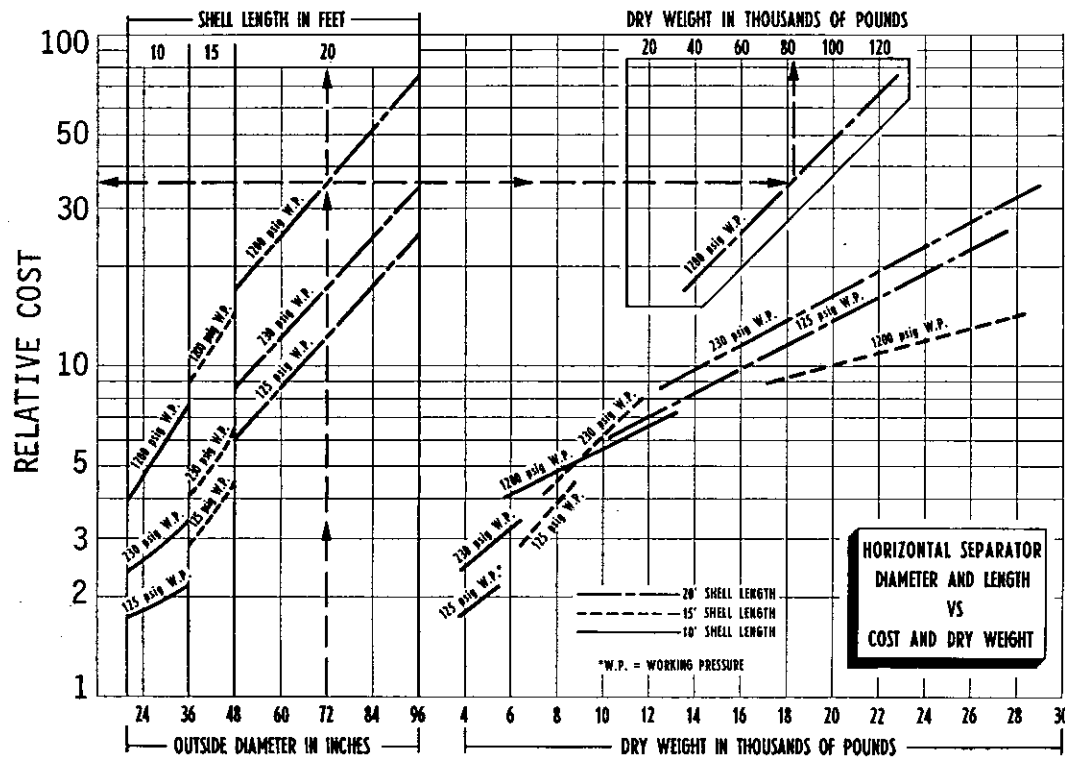
Surface Area and Weight of Vessels with 2:1 Ellipsoidal Heads

Multiply W by: 1.022 for stainless steel

1.135 for copper

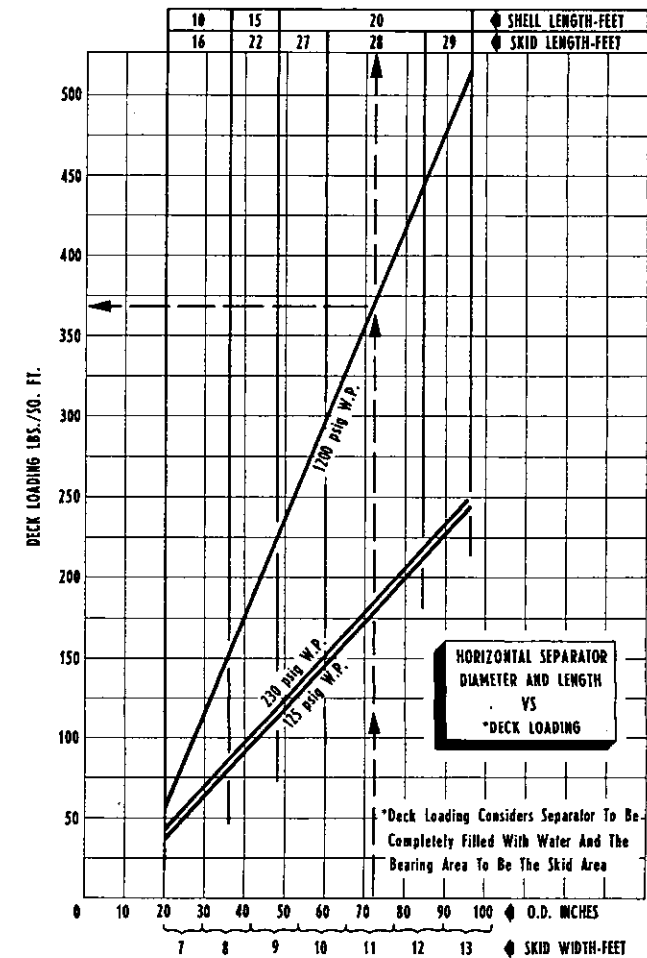
0.337 for aluminum

(Oil, Gas & Petrochem Equipment, May 1984)



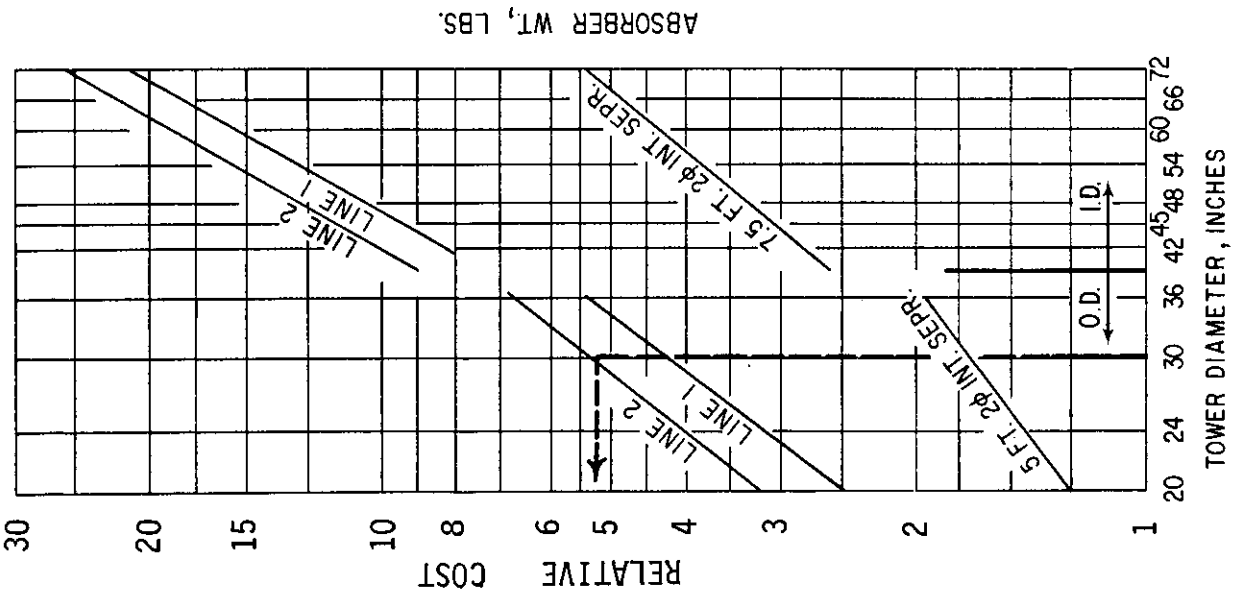
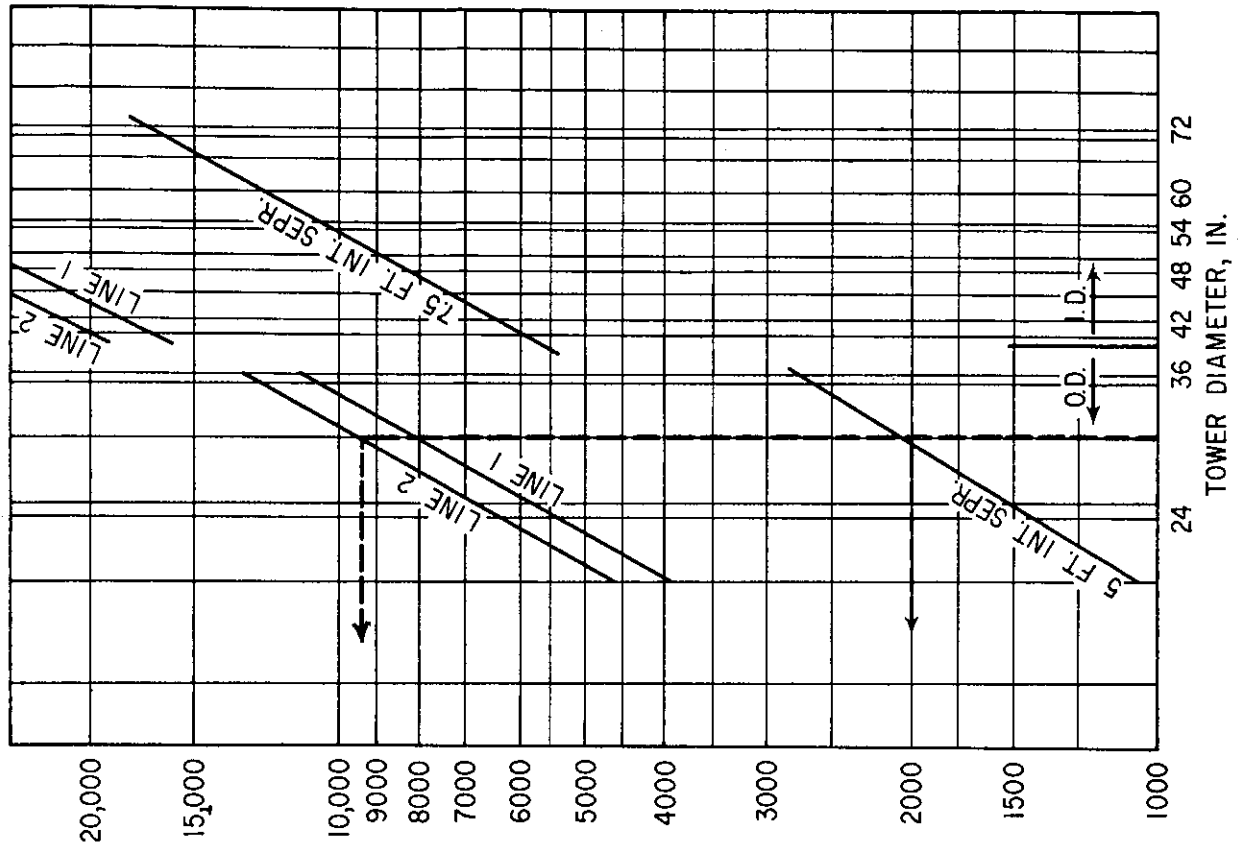
Relative Cost and Weight of Horizontal Separators

Cost and weight of various sizes of horizontal separators. Price includes skid-mounted ASME vessel complete with controls and external corrosion protection.

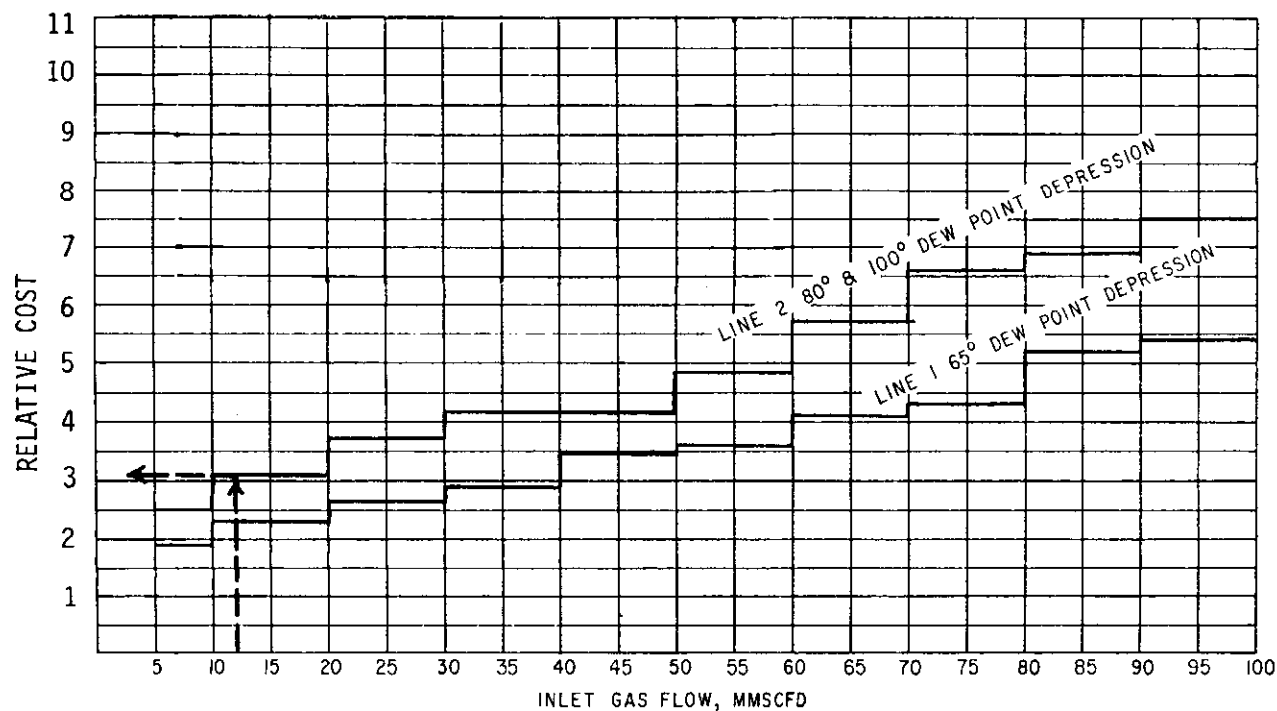


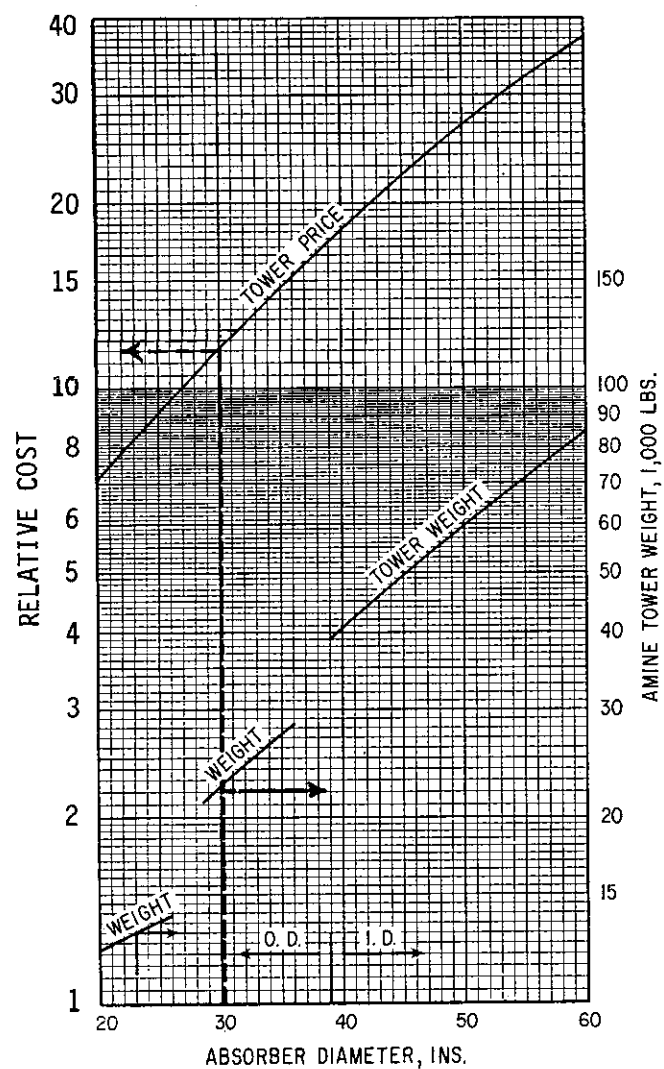
Relative Cost and Weight of Horizontal Separators

Deck loading characteristics and space required for horizontal separators. Units are assumed to be filled with water which can occur during testing, or if controls malfunction.

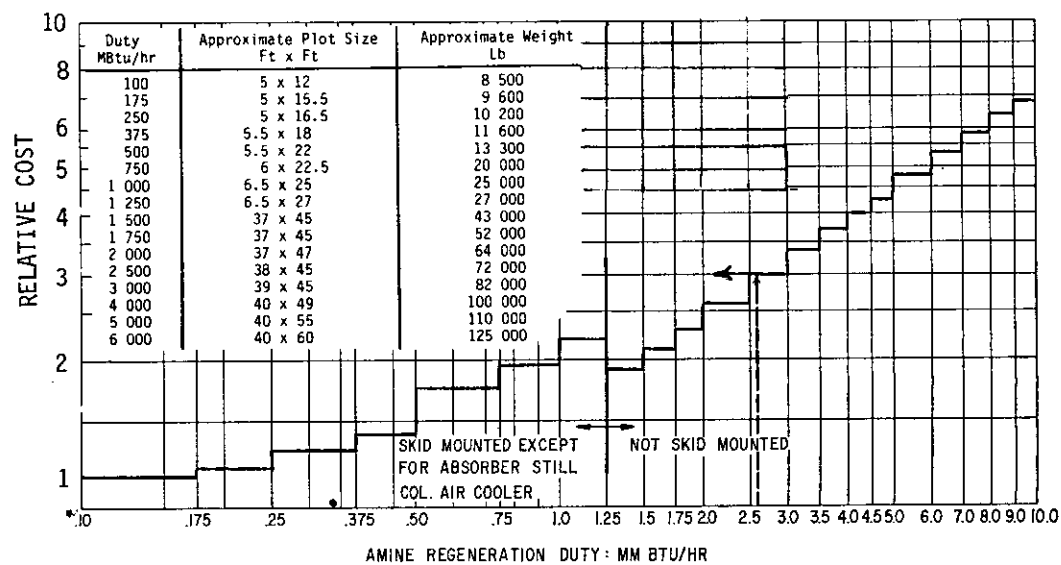


Approximate Deck Areas and Weights of Glycol Regeneration Units				
Inlet Gas Flow	Approximate Prot Size, Ft x Ft		Approximate Weights	
	65° Depression	80-100° Depression	65° Depression	80-100° Depression
5	4 x 6	4 x 8	2100	2700
10	4 x 9	4 x 10	2400	3400
20	5 x 12	5 x 14	3700	4900
30	6 x 14	6 x 16	4800	9500
40	6 x 16	6 x 19	6900	10 400
50	6 x 18	6.5 x 18	10 100	12 500
60	6 x 18	6.5 x 18	10 100	12 500
70	6.5 x 17	7 x 18	12 900	16 500
80	6.5 x 17	7 x 21	12 900	18 500
90	7 x 17	7 x 21	23 800	25 900
100	7 x 17	7 x 21	23 800	25 900

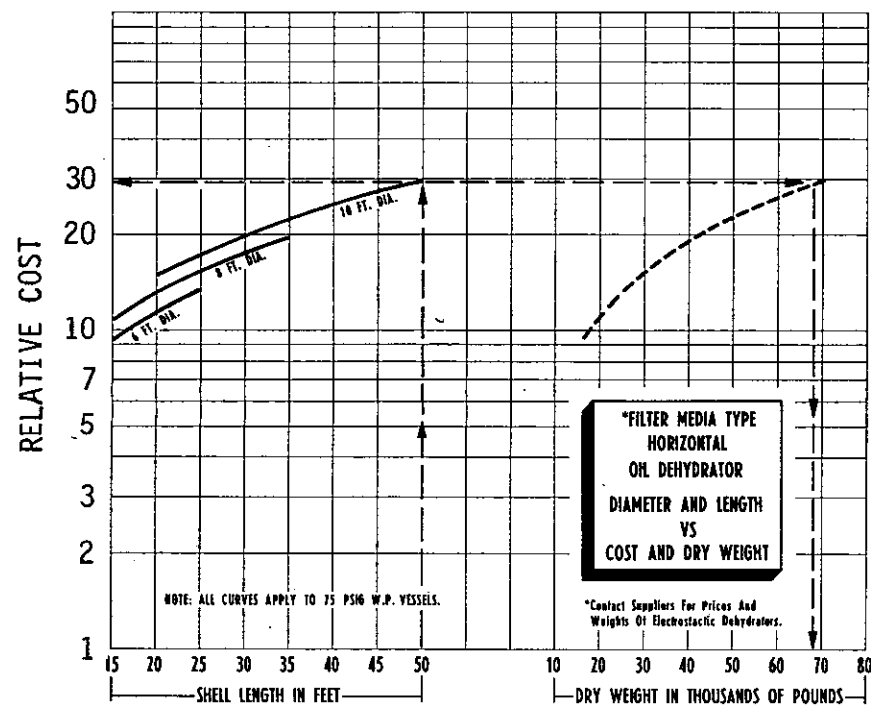




Relative Cost and Weight of an Amine Absorber

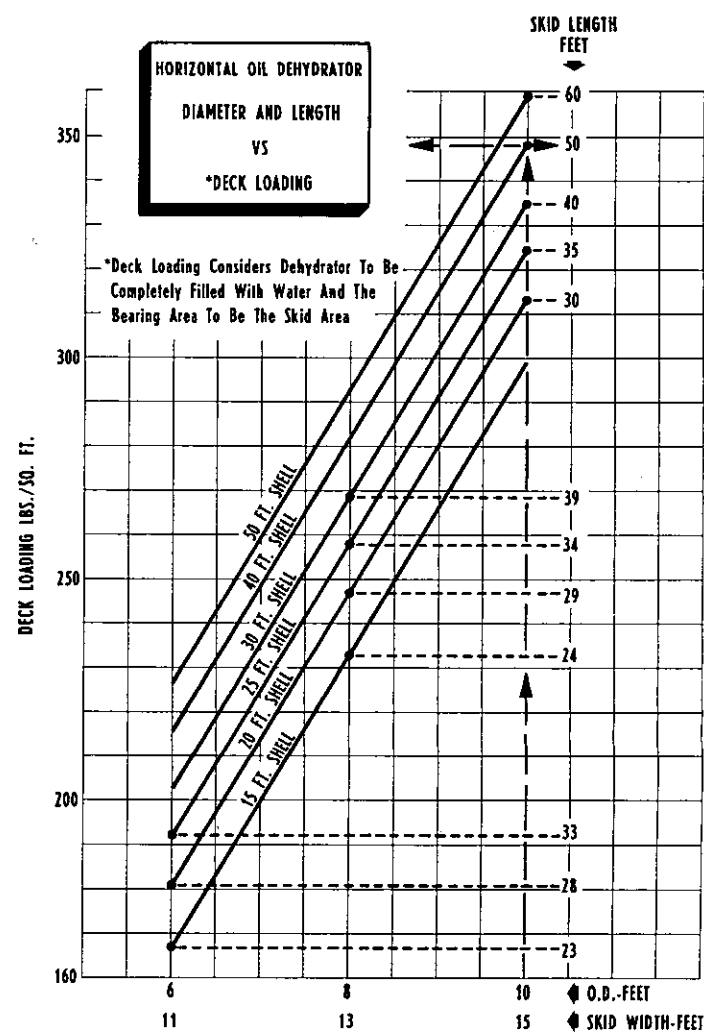


Relative Cost, Weight and Space Requirements of an Amine Regeneration Unit



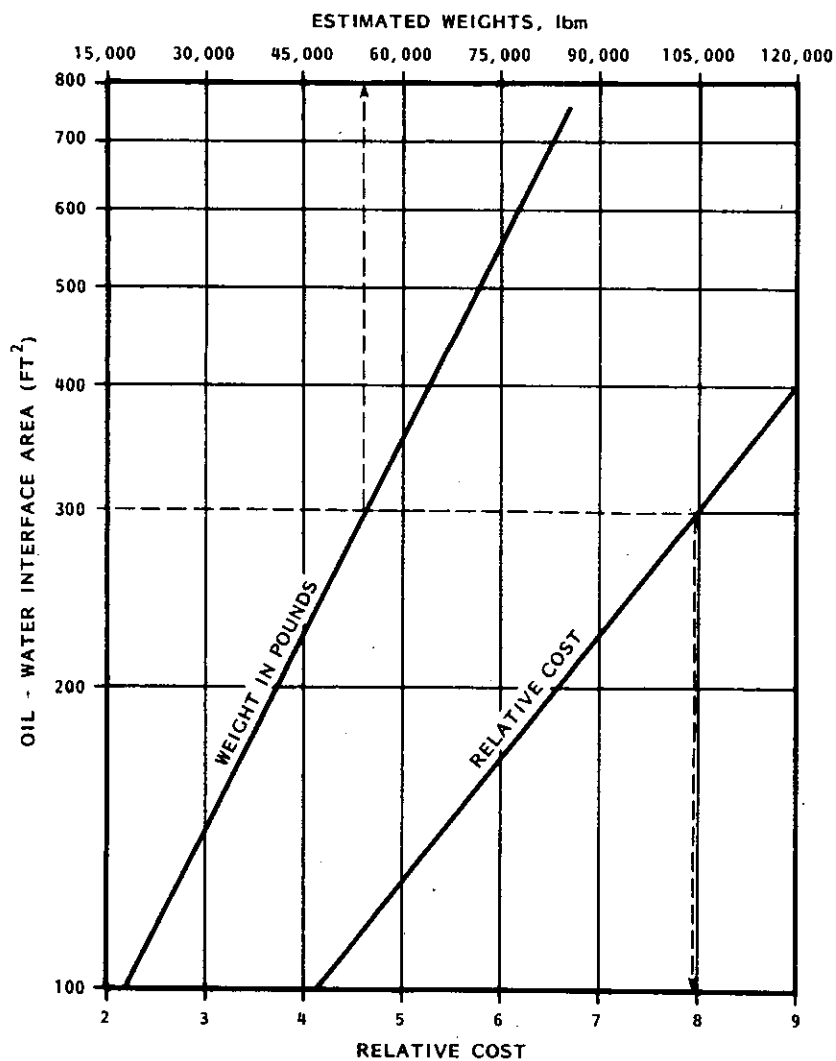
Relative Cost and Weight of Horizontal Oil Dehydrators

Cost and weight of various sizes of horizontal filter media type treaters. ASME vessels are skid-mounted, equipped with all controls and provided with external corrosion protection.

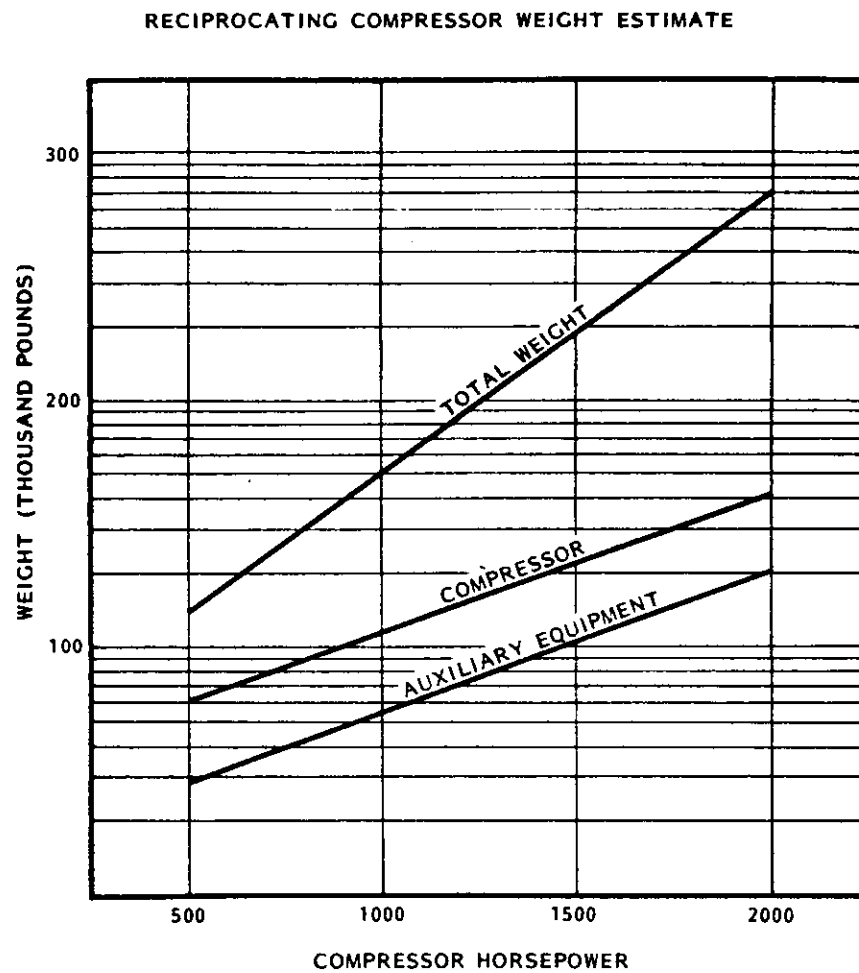


Relative Cost and Weight of Horizontal Oil Dehydrators

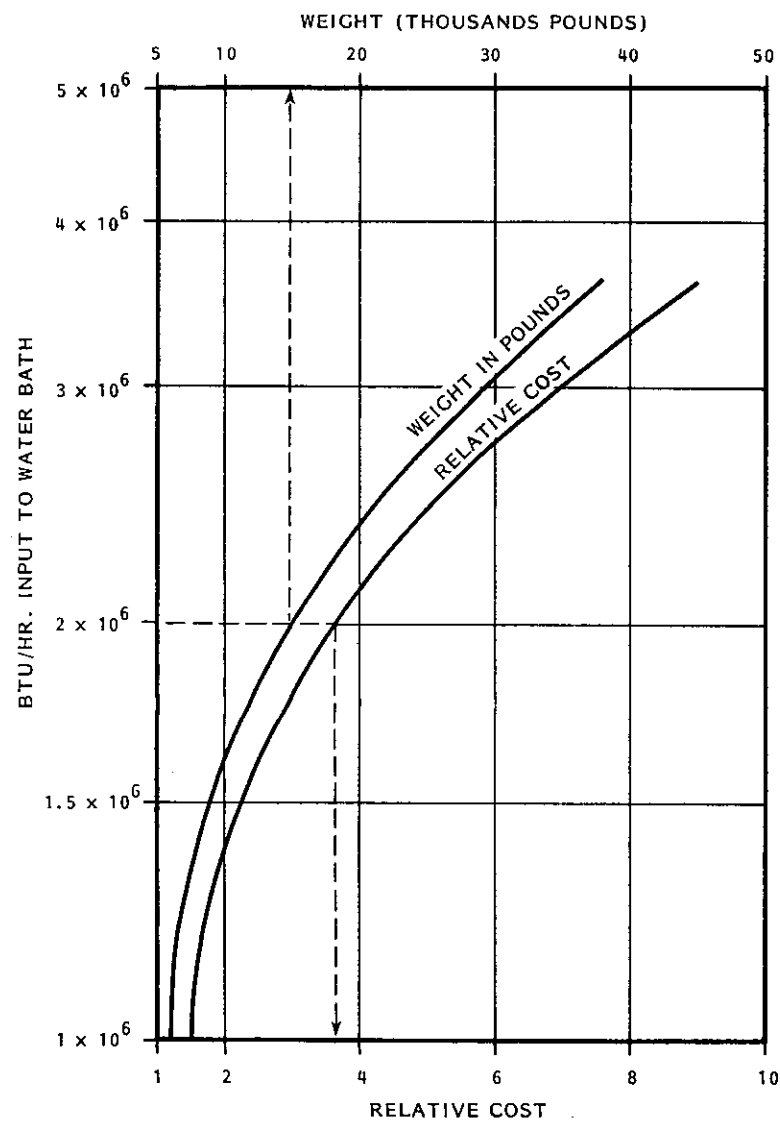
Space requirements and deck loading data for horizontal treaters. Information from this chart is helpful when locating vessels on platform foundations in marsh or water



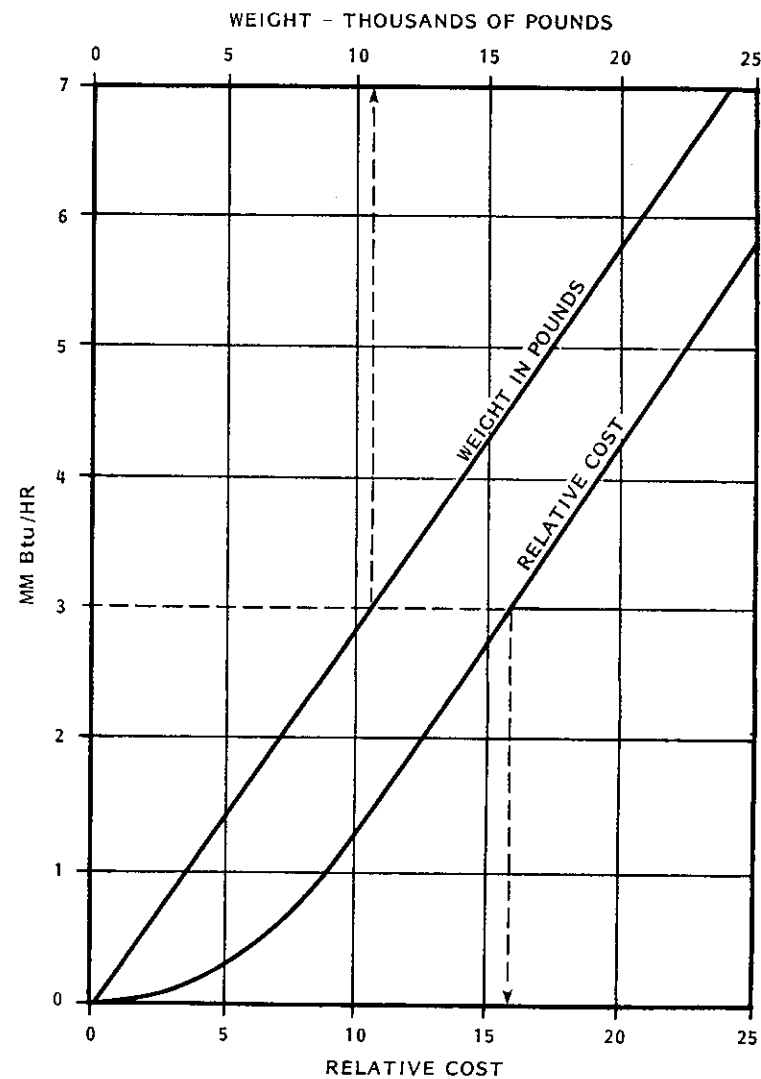
Estimated Costs and Weights for Crude Oil Dehydrators with Electrically Assisted Coalescence



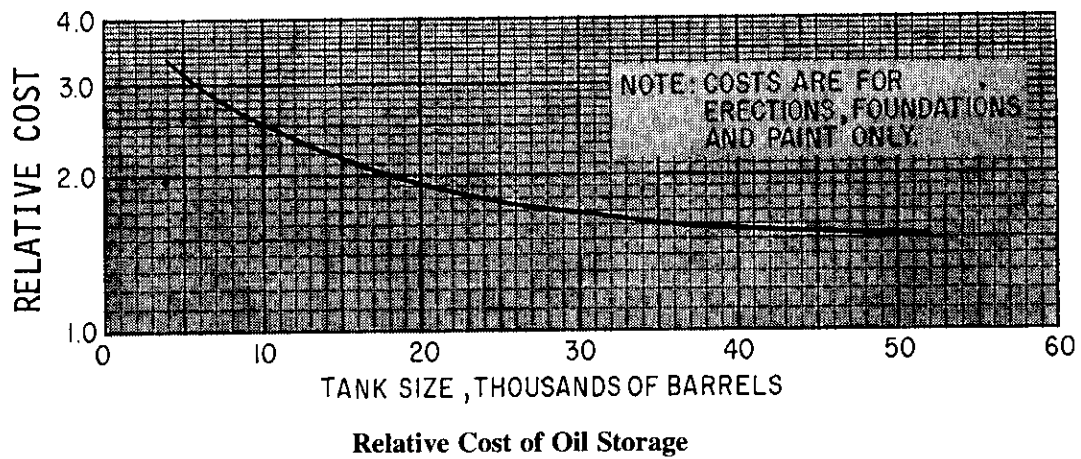
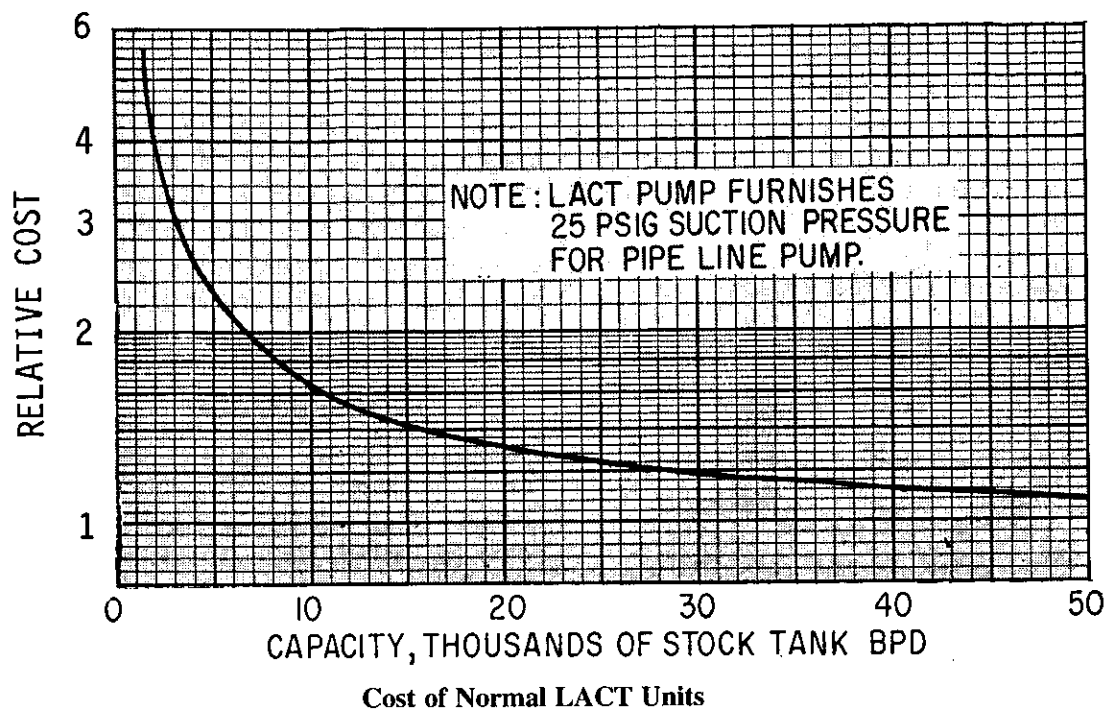
Reciprocating Compressor Weight Estimate



Weight and Cost of Indirect Heaters with Coil Operating at 500 psi



Estimated Costs and Weights of Direct-Fired Liquid Heaters, 25 psi Maximum Operating Pressure



NOTES:

12

FUNDAMENTALS OF RATE PROCESSES

Engineers are concerned about rate – the rate of heat transfer, the rate of doing work (power), the rate of flow, the rate of mass transfer, etc. The form of the rate equation used is the same for all engineering disciplines. The basic form is

$$\text{Rate} \propto \frac{\text{Driving Force}}{\text{Resisting Force}} \quad (12.1)$$

The proportionality sign may be replaced by an equal sign if we add a proportionality constant to the right-hand side of the equation.

$$\text{Rate} = \frac{(k)(\text{Driving Force})}{\text{Resisting Force}} \quad (12.2)$$

Constant "k" is the general proportionality constant. For each specific application it is given a name like heat transfer coefficient, mass transfer coefficient, etc.

Driving Force

The table below shows the common driving forces.

Process	Driving Force	Rate
Fluid flow – piping, vessels, porous media	Pressure	mass/unit time
Heat transfer – heating, cooling, mass transfer	Temperature	energy/unit time
Mass transfer – absorbers, fractionators, stabilizers	Concentration	mass/unit time
Electricity	Voltage	amperes

Some mass transfer processes like absorption and fractionation also involve heat transfer. A change of phase is involved. So ... a given piece of equipment must be designed so that both the mass and the equivalent energy change involved may be transferred between phases. This is why design demands both a heat and material balance.

Resisting Force

Two variables affecting resistance may be defined easily – length (L) and area (A). Resistance to flow is proportional to the distance of flow. The longer the flow path the greater is the resistance, on a relative basis.

Resistance is inversely proportional to area. The larger the area available for flow, the less the relative resistance will be per unit of flow.

Equation 12.2 may then be written generally as:

$$\text{Rate} = \left(\frac{k}{L} \right) (A) (\text{Driving Force}) \quad (12.3)$$

If no factors which affect resistance other than L or A are included in the working equation, the proportionality constant must contain in its numerical value the effect of all these factors.

In some cases L is not included because it is small and not easily, or accurately, measurable. If L is not shown in the equation it is included in the proportionality constant.

The proportionality constant must always be an empirical number. In a test one can measure rate, driving force, area, and L (if feasible). An equation of the form of 12.3 is then solved for the proportionality constant. The " k " thus found is applicable only to the system being tested. What is done is to make a series of controlled tests using various combinations of the known variables affecting the process. A correlation for " k " is then prepared.

So ... k is like Z , K , H , S and a large number of like quantities. Each is an empirical number which can be used reliably only if it was obtained on a system comparable to one on which it is applied. No matter how complex the equation, or how many numbers are used to the right of the decimal place, this is still true. This is why no one correlation can be used indiscriminately for all applications.

Some Examples of Linear Flow

Linear flow is flow in only one direction. If we further have only one resisting force, some very simple rate equations can be written which illustrate the rate equation concept.

Electricity

Basic equation:
$$I = \frac{E}{R}$$

Where: I = rate in amperes E = driving force in volts
 R = resistance in ohms

Note that the units have been defined so that " k " = 1.0.

Conduction of Heat

Basic equation:
$$Q = \left(\frac{k}{L} \right) (A) (\Delta T)$$

Where: Q = rate of heat transfer
 k = thermal conductivity (proportionality constant)
 L = length of flow
 A = area through which transfer occurs
 ΔT = temperature driving force

Heat Across a Thin Fluid Film

Basic equation:
$$Q = h A \Delta T$$

Where: h = film coefficient (proportionality constant)

The value of " h " depends on all factors which affect film thickness as well as any other factors affecting resistance.

Mass Transfer

Basic equation:

$$Q_m = k_G A (c_1 - c_2) \quad \text{or} \quad Q_m = k_{GA} (c_1 - c_2)$$

Where: Q_m = mass rate

k_G or k_{GA} = mass transfer coefficient

c_1 and c_2 = concentration change causing mass transfer

When area is not measurable directly it is incorporated into the coefficient k_{GA} . In some cases area is made a function of volume, as noted for packings in Chapter 8.

Porous Media Fluid Flow

Basic equation:
$$Q = \left(\frac{k}{\mu} \right) (A) \left(\frac{\Delta P}{L} \right)$$

Where: Q = mass flow rate

k = permeability (proportionality constant)

A = area through which flow occurs

μ = viscosity of fluid

ΔP = pressure driving force

L = length of flow section

Notice that viscosity has been added to the resisting forces. Permeability now represents all resistances except A , L and μ .

The above examples show how the rate equations are set up. We have shown them using "delta" notation. They also may be written in differential form and then integrated.

RESISTANCES IN SERIES

Each of the previous equations has been written for the calculation of rate across a single resistance. In a given system or piece of equipment there may be more than one resistance in series. Series flow is where the total flow passes through a sequence of resistances.

Let us illustrate this case using heat transfer between two fluids, across a solid surface.

There are three basic resistances in series – two fluid films and the solid wall separating the fluids. The fluid films are very thin so it is not practical to include "L" as a resistance.

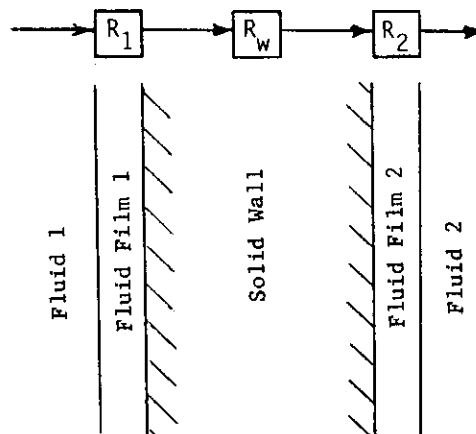
Total $R = R_1 + R_w + R_2$. But since the same amount of heat must flow through each resistance, $Q = Q_1 = Q_w = Q_2$. Or,

$$Q_1 = \Delta T_1 / R_1 = h_1 A_1 \Delta T_1$$

$$Q_w = \Delta T_w / R_w = (k_w / L_w) A_w \Delta T_w$$

$$Q_2 = \Delta T_2 / R_2 = h_2 A_2 \Delta T_2$$

So, $R_1 = 1/h_1 A_1$, $R_w = L_w/k_w A_w$, $R_2 = 1/h_2 A_2$



Now, $\Delta T = \Delta T_1 + \Delta T_w + \Delta T_2$. So, we can combine terms to arrive at a total equation

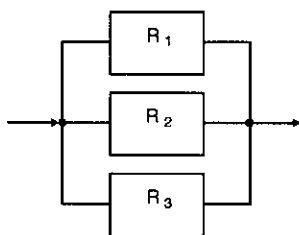
$$Q = \frac{\Delta T}{R} = \frac{\Delta T}{\frac{1}{h_1 A_1} + \frac{L_w}{k_w A_w} + \frac{1}{h_2 A_2}}$$

This equation will be used in Chapter 13. The equivalent for fluid flow will be used in Chapter 10.

RESISTANCES IN PARALLEL

If the resistances are in parallel the total flow rate is divided among these resistances. The flow through each resistance is proportional to that resistance. The driving force is the same across each parallel resistance.

To illustrate this, assume that the figure at left on the next page applies to an electrical circuit with three wires in parallel.



For this case

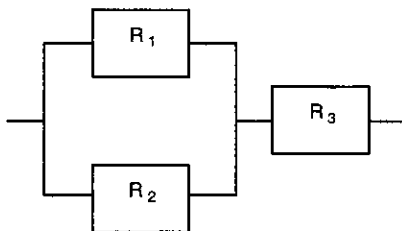
$$I = I_1 + I_2 + I_3 \quad , \quad E = E_1 = E_2 = E_3$$

$$\text{and} \quad , \quad I_1 = E/R_1 \quad , \quad I_2 = E/R_2 \quad , \quad I_3 = E/R_3$$

The same basic relationships apply for all flow systems containing resistances in parallel

COMPOUND RESISTANCES

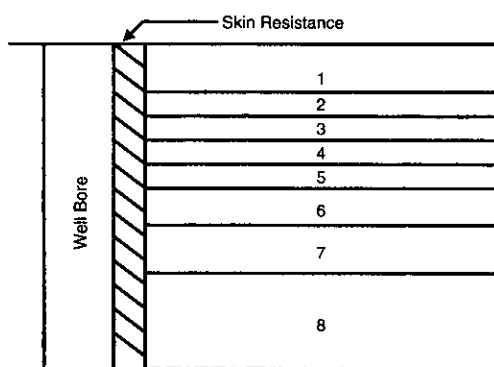
A given system may have resistances in both series and parallel. Consider the simple electrical system shown below.



$$I = I_1 + I_2 = I_3 \quad \text{and} \quad E = E_1 + E_3 = E_2 + E_3$$

$$E_1 = E_2$$

If one can calculate the value of R_1 and R_2 , the relative flow rate through each parallel line can be determined.



A similar example is found in the well bore. It is customary to divide a reservoir into a series of layers for calculation purposes. Each layer (1, 2, 3, etc.) has a given permeability and height assigned from logs and cores. The reservoir proper is therefore characterized as a series of parallel resistances.

In the process of drilling a reservoir some damage may result around the well bore. This "skin" is a resistance in series.

In Chapter 10, the looping of a line is another example of compound resistances.

GENERAL APPLICATIONS

The rate equations, when corrected for the effect of resistance patterns, serve as a foundation for performance along with thermodynamic concepts, equilibrium, phase behavior and other principles discussed in the first ten chapters. Keep these things in mind as we begin consideration of specific equipment and processes.

In absorption and fractionation, where vapor and liquid phases are involved, the rate of transfer of any component from one phase to another is governed by its relative concentration in the respective phases. If its concentration is less in the vapor phase than in the liquid phase, the component will vaporize at some rate. With reverse relative concentration, condensation will occur.

At low pressures, concentration is often expressed in terms of partial pressures. At higher pressures, outside the ideal gas range, partial fugacities may be used. As noted elsewhere, concentration may be expressed as a mole fraction or any other unit that makes the calculation convenient.

The area of contact between phases is a critical variable that may be controlled to some degree. In adsorption processes the active surface area of the solid is a factor. Since most of this area occurs in the small capillaries that honeycomb the solid, capillary size can also be a factor. Obviously, a molecule larger than the capillary diameter will be unable to even get to the internal surface and effective mass transfer would be negligible. As capillary size increases, the surface area per unit volume of adsorbent decreases. Consequently, in some adsorption processes, the effective area available is limited by the size of molecules being processed.

The rate at which a molecule can move down a capillary large enough to accommodate it will depend on a number of factors such as wettability of the fluid, diffusion in the fluid phase, etc. Such variables are necessarily included in the mass transfer coefficient since they are not formally shown in the equation.

In the transfer of mass between vapor and liquid, the area of contact may be increased by breaking up the mass of gas or liquids into droplets – the smaller the droplet, the greater the contact area per unit of mass. A number of mechanical devices have been developed for this purpose. The velocity of the gas flowing upward through the contacting device is used to agitate the liquid into a froth. By restricting the gas flow, it passes through the liquid as smaller bubbles. This combination action supplies the area of contact. The amount of frothing, as controlled by gas velocity, is limited by the fact that entrainment of such liquid into the gas stream reduces the effective contactor efficiency. The combination of gas velocity and liquid droplet size must be such that the liquid will drop back into the mass of liquid from whence it came.

In quiescent contact between vapor and liquid the area of contact is very limited. Furthermore, the concentration of molecules is not uniform throughout each phase. Contacting the gas with the liquid as small bubbles not only increases surface area but also induces turbulence. The latter promotes better diffusion throughout the liquid phase.

Area of contact may also be increased by flowing the liquid along a large surface, in a thin sheet, countercurrent to the gas. The use of columns packed with ceramic or metal objects having a large surface area per unit volume is commonly done to achieve this.

Regardless of the mechanical device used to promote a large area of contact, the design of a mass transfer unit will always involve at least some elements of the following:

1. Assurance that the design is compatible with the conservation of mass and energy as represented by the first law of thermodynamics.
2. Some method for predicting the rate of transfer as a function of driving and resisting forces.
3. Knowledge of the equilibrium relationships applicable.
4. Fixing the allowable rates of fluid through the contacting device.

Design of fractionators and absorbers involves all four factors.

Finally, there is a philosophical reason for discussing the basic rate equation. Engineers tend to "hide" in their chosen areas of expertise and fail to "seek out" knowledge in other areas or perform in them. This is neither self-serving nor fulfilling the employer's needs. Principles are not bounded by geography, language or disciplinary areas. They apply to all systems. A basic understanding of the principles enables one to interact in a meaningful manner with "experts" in other areas – make suggestions, review work and generally communicate, all necessary ingredients in project development. One of my goals has been to remove fear of the unknown in the hope it will encourage you to continue to expand your technological horizons.

13

HEAT TRANSFER

The transfer of heat is necessary for control of: (1) a fluid temperature and/or its composition and phase; (2) the rate of mass transfer between phases; (3) the rate of chemical reactions and (4) suitable temperatures to prevent failure or reduced service life of the equipment. Provision for heat transfer is incorporated into most equipment. However, this chapter is devoted only to that equipment whose primary purpose is transfer of heat.

Heat transfer equipment can be divided into the following basic types.

Heaters:	Direct and Indirect
Fluid-Fluid:	Pipe-in-Pipe, Shell-and-Tube, Plate, Coil and Special Types for Specific Services
Coolers Utilizing Air:	Straight Aerial, Cooling Towers, Combination Air-Water

Each will be discussed to some degree in this chapter. The emphasis is on fluid-fluid types.

HEAT TRANSFER MECHANISMS

By definition, heat is that energy transferred solely as a result of a temperature difference, that is independent of mass transfer. There are three mechanisms of heat transfer – *conduction*, *convection* and *radiation*.

Conduction of heat occurs by the excitation of adjacent molecules where said molecules have little or no movement. Conduction thus is the primary mechanism in solids and may be an important component mechanism with some liquids at low flow rates.

Convection is that mechanism where heat energy is transferred by the physical movement of molecules from place to place. Any factor which enhances or hinders this movement affects the rate of heat transfer by convection. In most commercial fluid-fluid exchangers, convection is the most important mechanism.

As discussed in Chapter 12 and later herein, the usual heat transfer process is governed by a group of resistances in series. There are two fluid films governed primarily by convection, the solid separating the fluid governed by conduction and possibly some other corrosion, scale or deposition films also governed by conduction.

Radiation is the process whereby a body emits *heat waves* that may be absorbed, reflected or transmitted through a colder body. The sun heats the earth by means of electromagnetic waves. A hot body emits a

whole spectrum of wave lengths. Radiation which affects the eye as light extends roughly from 0.00004-0.00008 cm in wave length. To the right of this visual spectrum is the *infrared* region; to the left is the *ultraviolet* region. Heat is transferred throughout the full wave length range. As temperature increases the predominant wave lengths become shorter. A detailed understanding of radiation is provided by the *quantum theory* of physics.

BASIC CONDUCTION/CONVECTION EQUATIONS

In most fluid-fluid exchangers the temperatures are not high enough for radiation to be a significant mechanism. Since the coefficients used to calculate performance are empirical, they incorporate any radiant effects that might have been present in the test system.

As noted in Chapter 12, one may calculate the heat transfer process by the equation:

$$Q = Q_1 = Q_w = Q_2 = h_1 A_1 \Delta t_1 = (k/L) A_w \Delta t_w = h_2 A_2 \Delta t_2 \quad (13.1)$$

The values of "h" are proportionality constants used to characterize the liquid film resistance determined from experimental data or general correlations. The value "k" is the thermal conductivity of the solid separating the two fluids – a measurable property of that solid.

It is convenient to show total heat transfer per unit time in terms of an overall heat transfer coefficient "U."

$$Q = U_1 A_1 \Delta t_m = U_2 A_2 \Delta t_m \quad (13.2)$$

The overall coefficient "U" is related to the film coefficients and thermal conductivity by the equations:

$$U_1 = \frac{1}{\frac{1}{h_1} + \frac{A_1 L}{k A_w} + \frac{A_1}{h_2 A_2} + F_f} \quad (13.3)$$

$$U_2 = \frac{1}{\frac{A_2}{h_1 A_1} + \frac{A_2 L}{k A_w} + \frac{1}{h_2} + F_f} \quad (13.4)$$

Where:

- h_1 = film coefficient for fluid 1
- h_2 = film coefficient for fluid 2
- k = thermal conductivity of solid wall
- A_1 = surface area of pipe for film h_1
- A_2 = surface area of pipe for film h_2
- A_w = average wall area of pipe or tubing
- L = wall thickness of pipe or tubing
- F_f = fouling factor

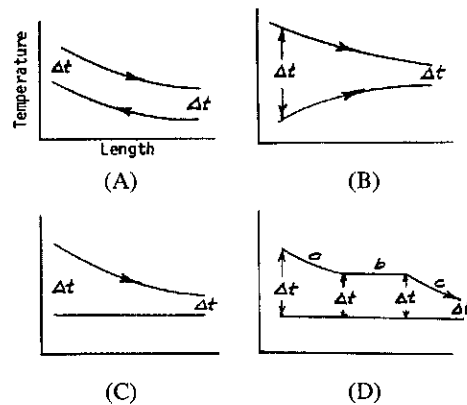
The fouling factor (F_f) accounts for scale, rust, and the like which form on the surface with use and in effect provide an additional resistance to heat flow. The fouling factor will vary widely with conditions.

Values of the overall coefficient "U" may be predicted from Equations 13.3 or 13.4 or from actual performance. Most heat exchanger quotations show the overall "U" used in preparing them. These, plus plant operating data, are a valuable source of information for future planning. Equations 13.3 and 13.4 differ only in the surface area used for reference. They assume one fluid is flowing inside of the exchanger tubing and one is flowing along the outside surface. "U" will vary with area so that $U_1 A_1 = U_2 A_2$. In shell-and-tube exchangers the heat transfer area "A" is almost always based on outside tube wall area.

EFFECTIVE ΔT

Equation 13.2 is the basic equation used for design. It contains the term Δt_m . This is the mean Δt because the Δt across the wall surface varies with location as shown below.

- (A) Two fluids flowing countercurrent, no phase change.
- (B) Two fluids flowing concurrent, no phase change.
- (C) One fluid flowing and one boiling (or condensing).
- (D) Superheated vapor being cooled to saturation (a) condensing (b) and being subcooled as liquid. The other fluid is boiling or condensing.



The only temperatures that we can measure conveniently are at the inlet and outlet ends of the exchanger. Thus, we can measure two Δt 's. The larger we will call Δt_1 , the smaller Δt_2 . Δt_2 is also called the *approach*. It designates how close the temperatures of the two fluids approach each other in the exchanger.

In concurrent flow the fluids flow in the same direction. In countercurrent flow they flow in opposite directions. Most exchangers use countercurrent flow, or as close to it as possible, since it is more efficient.

The basic equation for estimating Δt_m is:

$$\Delta t_m = (F) \left[\frac{\Delta t_1 - \Delta t_2}{\ln (\Delta t_1 / \Delta t_2)} \right] \quad (13.5)$$

- Where: Δt_m = log mean temperature difference (LMTD)
 F = factor for heat exchanger
 Δt_1 = largest Δt (at one end of the heat exchanger)
 Δt_2 = smallest Δt (at one end of the heat exchanger)
 \ln = logarithm to the base e

The value of F depends on the geometry of the fluid flow in the exchanger and will be discussed later for each type. $F = 1.0$ for a concentric pipe-in-pipe exchanger. Equation 13.5 can be derived from the calculus for this situation.

Approach

The approach, Δt_2 , is an economic choice. Its specification governs heat exchanger cost. As Δt_2 gets smaller, LMTD becomes smaller and area required becomes larger. As LMTD approaches zero, area approaches infinity. Since the cost of the heat exchanger is a direct function of area, specification of approach has a direct effect on cost. In order to have the optimum cost installation, a series of exchangers in series may be used.

The approach used often will be in the following range:

- Aerial coolers, 10-25°C [18-45°F]
- Water cooling of hydrocarbon liquids and gases, 8-12°C [14-22°F]
- Liquid-liquid heat exchange, 11-25°C [20-45°F]
- Refrigeration chillers on gas-liquid streams, 4-6°C [7-11°F]

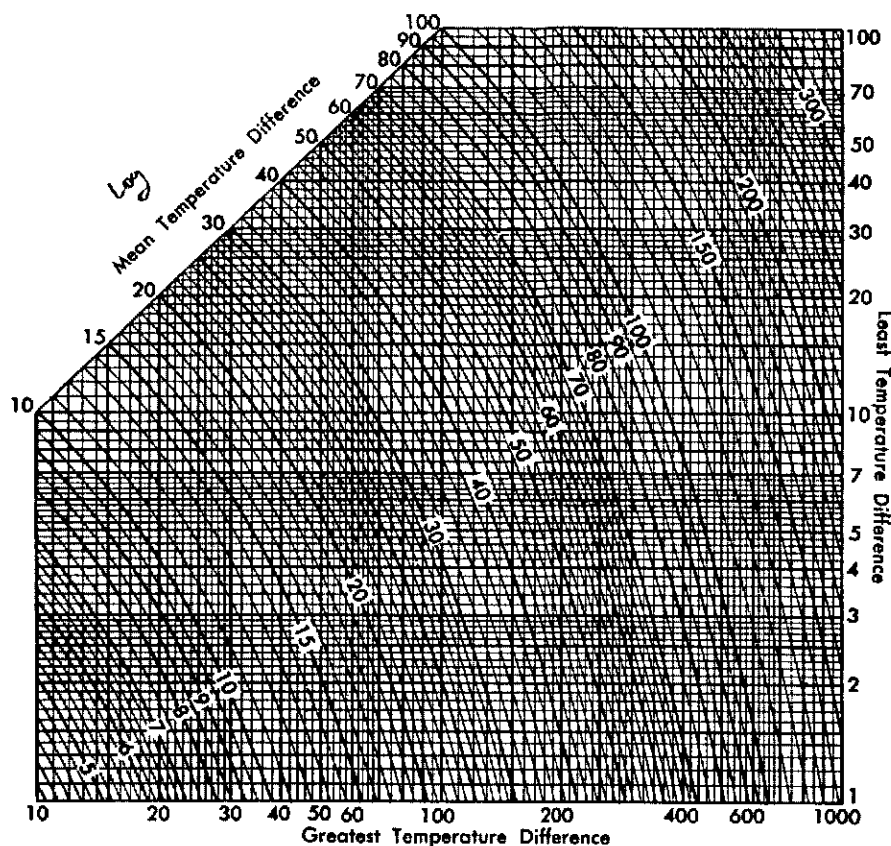
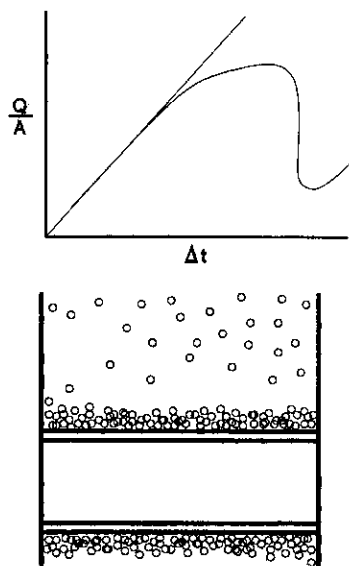


Figure 13.1 Nomograph for LMTD

When specifying heat exchangers it often is desirable to specify a maximum or minimum approach to the vendor. This does not fix the actual approach. It merely establishes an upper or lower limit, below or above which the actual approach must occur.

Vaporizing (Boiling) Liquids

There is a special concern when one of the heat exchanger fluids is vaporizing. This occurs in refrigeration chillers and fractionation reboilers, as two examples.



From Equation 13.2, you would expect a plot of Q/A versus Δt to yield a line like the dashed line at left. It does except for boiling liquids. The solid curve at right is what really occurs with boiling liquids. At some value of Δt the curve changes direction and Q/A decreases rapidly to a minimum, after which it begins to rise again. Why?

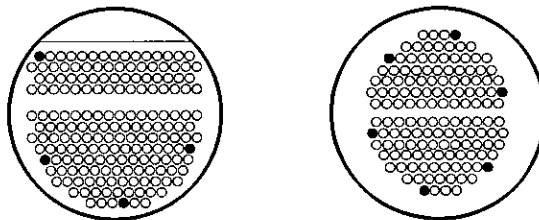
As shown in the sketch at left, a layer of gas bubbles can build up around a tube if vaporization occurs at the tube wall faster than the vapor can disengage and rise through the liquid. This layer of bubbles forms an extra resistance in series and is a type of fouling factor.

When Δt across the tube reaches a critical point, the bubble layer forms and Q/A decreases. If Δt continues to increase, the layer resistance stabilizes and Q/A begins to increase again.

The critical Δt depends on the liquid and the character of the tube surface. The critical Δt may occur as low as 20-35°C. Special tube surfaces are marketed which are designed to minimize bubble layer formation.

There are two basic mechanical factors which affect vapor disengagement – spacing and arrangement of the exchanger, and the area available between the liquid and vapor phases. As vapor forms it must get away from the surface quickly. There also must be enough surface area so that the resistance at the vapor-liquid interface does not limit vapor disengagement.

In the figure at right are shown two tube configurations among the many available. This is known as *triangular layout* since the tubes in adjacent rows are not directly above or below each other. To improve vapor disengagement between tubes, the tube pitch is typically 1.5 to 2 times the tube diameter.



Another alternative is the *square layout* where tubes in adjacent rows are directly above or below each other. Although not as common as triangular layout, square layout has been used in corrosive service such as amine regeneration.

Notice in the previous figure that there is room above the tubes for vapor. The arrangement shown is typical for chillers and reboilers where the liquid covers all tubes. The area of the liquid surface must be sufficient, which means that the shell diameter must be larger than that needed to merely hold the tubes. The arrangement at the right might be used for a condenser to allow good vapor distribution. We have not shown baffles which might also be needed for good distribution.

As the tubes are farther apart there is more room for vapor to rise. But, the cost of the exchanger increases. Be sure the low bid on your reboiler or chiller has enough vapor space.

Also, be sure the vapor outlet flanges and piping have sufficient area. If not, vapor can back up, "choke" the exchanger, and limit capacity even though the tube area is adequate.

BASIC HEAT BALANCE

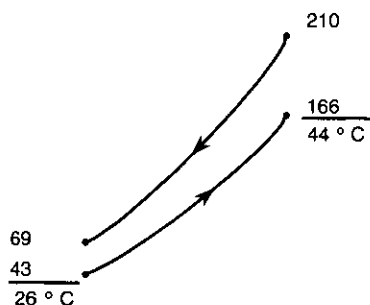
In a fluid-fluid heat exchanger the overall balance reduces to $\Delta H = Q$. Furthermore, the ΔH of one fluid equals the ΔH of the other fluid if one ignores heat losses to, or heat gains from, the atmospheric air. Heat loss or gain is normally considered to be zero in exchanger heat balances. If the vessel is properly insulated the heat loss or gain should be less than 5% of Q . This is compensated for when choosing the actual equipment.

In solving a balance like that discussed previously, two flow rates and three temperatures may be specified. In this case, the fourth temperature must be calculated. This is found by equating the ΔH equations for the two fluids and solving for the unknown temperature.

If one needs to solve Equation 13.5 for one Δt , knowing Δt_m and the other Δt , a trial-and-error calculation results. This equation may be approximated by Equation 13.6.

$$2 (\Delta t_m)^{1/3} = (\Delta t_1)^{1/3} + (\Delta t_2)^{1/3} \quad (13.6)$$

Example 13.1: Lean oil is to be cooled from 210°C to 69°C by rich oil entering the HEX at 43°C and leaving at 166°C. The rich oil is flowing inside the tubes of a pipe-in-pipe exchanger. Calculate Δt_m .



$$\Delta t_1 = 44^\circ\text{C} \quad , \quad \Delta t_2 = 26^\circ\text{C}$$

For a pipe-in-pipe exchanger, $F = 1.0$

From Equation 13.5, or Figure 13.1

$$\Delta t_m = 1.0 \left[\frac{44 - 26}{\ln (44/26)} \right] = (1.0)(34.2) = \underline{34.2^\circ\text{C}}$$

In this lean oil/rich oil exchanger the flow rate of lean oil is 20 000 kg/h and its specific heat is 2.30 kJ/kg·°C. Calculate area.

$$\Delta H (\text{lean oil}) = Q = (20\,000)(2.30)(210 - 69) = 6\,486\,000 \text{ kJ/h} = 1802 \text{ kW} = \Delta H (\text{rich oil})$$

If a pipe-in-pipe exchanger is used, $F = 1.0$, and if $U = 466 \text{ W/m}^2\cdot^\circ\text{C}$,

$$A = \frac{1802 \times 10^3}{(466)(34)} = \underline{113 \text{ m}^2}$$

From the heat balance one could calculate the flow rate of rich oil that would produce the temperatures specified. Any change in the properties or flow rate of one stream will change its ΔH which, in turn, changes Q and thus the LMTD.

HEAT UNITS

The rate of heat transfer (Q) normally is expressed in kJ/h, kW or Btu/h. Since enthalpy usually is expressed in kJ/kg, the use of kJ/h is convenient even though the watt is the basic metric unit for rate of energy transfer.

$$1 \text{ Btu/h} = 1.055 \text{ kJ/h} = 0.293 \text{ W} \quad , \quad 1 \text{ kJ/h} = 0.278 \text{ W}$$

$$1 \text{ kW} = 3413 \text{ Btu/h} = 3600 \text{ kJ/h}$$

Overall and film coefficients (U and h) are proportionality constants whose units depend on the rest of the equation.

$$1 \text{ Btu}/(\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}) = 20.42 \text{ kJ}/(\text{h}\cdot\text{m}^2\cdot\text{K}) = 5.678 \text{ W}/(\text{m}^2\cdot\text{K})$$

Thermal conductivity (k) is also a proportionality constant but is expressed per unit length of the conduction flow path.

$$1 \text{ (Btu}\cdot\text{ft)} / (\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}) = 6.23 \text{ (kJ}\cdot\text{m)} / (\text{h}\cdot\text{m}^2\cdot\text{K}) = 1.73 \text{ W}/(\text{m}\cdot\text{K})$$

$$1 \text{ (Btu in.}/(\text{h}\cdot\text{ft}^2\cdot^\circ\text{F}) = 0.52 \text{ (kJ}\cdot\text{m)} / (\text{h}\cdot\text{m}^2\cdot\text{K}) = 0.144 \text{ W}/(\text{m}\cdot\text{K})$$

DETERMINATION OF EXCHANGER PERFORMANCE

The performance of a heat exchanger, regardless of type, is governed by Equations 13.2-13.4. In most instances Q is fixed by an energy balance. Three of the temperatures are specified and the fourth is calculated from an energy balance. Area (A) thus depends on the overall heat transfer coefficient used.

The best source of " U " values is from actual tests on comparable equipment. Continued tests over a period of time build up a data base for various conditions of operation, fluids and types of exchangers. A second good source of " U " values is data from reliable vendors.

Figure 13.2 is a general summary of " U " values for shell-and-tube type exchangers. The upper nomograph is for exchangers with only nominal fouling that operate at acceptable fluid velocity levels.^(13.1) The lower table is another compilation of such data. Correlations like these are not recommended for design calculations but are useful for general estimation of behavior and/or the evaluation of alternative systems.

Calculation of " U " from Equation 13.3 or 13.4 may be necessary. If so, values of " k " and " h " are required.

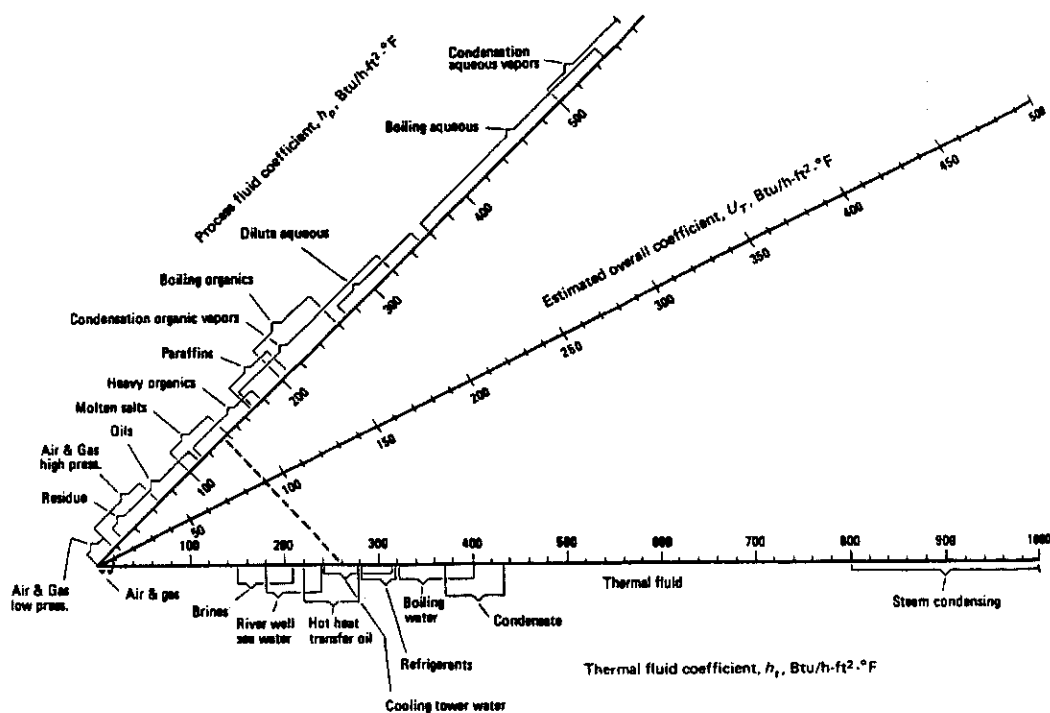
Example 13.2: High pressure natural gas is to be cooled with cooling water in a compressor after cooler. The outside tube diameter is 19 mm [0.75 in.] and the inside diameter is 14.8 mm [0.584 in.]. The estimated film coefficients are $454 \text{ W/m}^2\cdot\text{K}$ [$80 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$] for the gas and $1703 \text{ W/m}^2\cdot\text{K}$ [$300 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$] for the cooling water. The thermal conductivity of the tube wall is $52 \text{ W/m}\cdot\text{K}$ [$30 \text{ Btu/h}\cdot\text{ft}\cdot^\circ\text{F}$]. The fouling factor on both the gas and water side is estimated to be $0.00018 \text{ m}^2\cdot\text{K/W}$ [$0.001 \text{ ft}^2\cdot\text{h/Btu}$]. Estimate the overall heat transfer coefficient. Assume natural gas is on the tube side.

$$A_o = 0.06 \text{ m}^2/\text{m} [0.196 \text{ ft}^2/\text{ft}] \quad A_i = 0.0465 \text{ m}^2/\text{m} [0.153 \text{ ft}^2/\text{ft}]$$

$$A_w = 0.0529 \text{ m}^2/\text{m} [0.174 \text{ ft}^2/\text{ft}] \quad L = 0.0042 \text{ m} [0.0138 \text{ ft}]$$

$$\begin{aligned} \frac{1}{U} &= \frac{1}{h_o} + \frac{A_o L}{A_w k} + \frac{A_o}{A_i h_i} + f_i + f_o \\ &= \frac{1}{1703} + \frac{(0.06)(0.0042)}{(0.0529)(52)} + \frac{0.06}{(0.0465)(454)} + 0.00018 + 0.00018 \\ &= 0.00388 \quad U = 258 \text{ W/m}^2\cdot\text{K} \end{aligned}$$

Similarly, for English units $U = 45.4 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$



Shell and Tube Exchangers		
Service	Coefficient	
	$\text{Btu}/(\text{hr-ft}^2\text{-}^\circ\text{F})$	$\text{W}/\text{m}^2\text{-K}$
Water Coolers		
Gas (to 3.5 MPa)	35-50	200-285
Gas (3.5-7.0 MPa)	50-80	285-455
Gas (over 7.0 MPa)	80-100	455-570
Natural Gasoline	70-90	395-510
MEA	130-150	740-850
Air	15-25	85-140
Water	170-200	965-1135
Water Condensers		
Amine regenerator	100-110	570-625
Fractionator overhead	70-80	395-455
Light hydrocarbons	85-135	480-765
Reboilers		
Steam	140-160	795-910
Hot oil	90-120	510-680
Glycol	10-20	55-115
Amine	100-120	570-680
General		
Oil-oil	80-100	455-570
Propane-propane	100-130	570-740
Rich MEA-lean MEA	120-130	680-740
Gas-gas (to 3.5 MPa)	50-70	285-395
Gas-gas (about 70 MPa)	55-75	310-425
Gas-propane chiller	60-90	340-510

Figure 13.2 Typical Values of Overall Heat Transfer Coefficients

Thermal Conductivity of Solids

The table following shows the thermal conductivity of common materials at about 93°C [200°F].

Material	Thermal Conductivity - k	
	W/(m·K)	Btu/(hr-ft-°F)
Copper	386	223
Aluminum	173	100
Admiralty Brass	121	70
Mild Steel	43	25
Silicon Bronze	26	15
Stainless Steel (18 Cr-8 Ni)	14	8
Inconel	14	8
90-10 Cu-Ni	52	30
70-30 Cu-Ni	31	18
Monel	26	15
Titanium	17	10

The choice of metal depends on thermal conductivity, cost, temperature of operation and the presence of corrosion/erosion problems. With sea water cooling, for example, the use of titanium may be indicated to minimize chloride cracking, even though it possesses low thermal conductivity and is more expensive to buy and fabricate.

Fluid Film Coefficients

The basic equation for calculating fluid films is obtained by dimensionless analysis.^(13.3) The correlation of data shows a proportionality relationship among the Nusselt, Reynolds and Prandtl dimensionless numbers. In a solvable form this relationship can be written as follows.

$$h = A \frac{k}{d} \left(\frac{d v \rho}{\mu} \right)^a \left(\frac{C_p \mu}{k} \right)^b \quad (13.7)$$

- Where:
- h = film coefficient, W/(m²·K) [Btu/(h-ft²-°R)]
 - A = proportionality constant
 - k = fluid thermal conductivity, W/(m·K) [Btu/(h-ft-°R)]
 - d = effective diameter, m [ft]
 - v = fluid velocity, m/s [ft/h]
 - ρ = fluid density, kg/m³ [lbm/ft³]
 - C_p = fluid specific heat, kJ/(kg·K) [Btu/(lbm-°R)]
 - μ = fluid viscosity, kg/(m·s) [lbm/(ft·h)]
 - a = coefficient on Reynolds No.
 - b = coefficient on Prandtl No.

In Equation 13.7 the effective diameter depends on the circumstances:

Tube inside diameter for fluid flowing inside tube

Tube outside diameter for fluid flowing normal to one or more tubes

For annular flow between concentric pipes use an equivalent diameter defined by the equation

$$d_e = \frac{d_2^2 - d_1^2}{d_1} \quad (13.8)$$

Where: d_e = equivalent diameter
 d_2 = inside diameter of outside pipe
 d_1 = outside diameter of inside pipe

The velocity term is the volumetric flow rate divided by the cross-sectional area of flow. When flow is normal to a series of tubes, the cross-sectional area is the area available for flow between the first row of tubes.

Equation 13.7 applies only for Newtonian fluids where viscosity is a proper measure of the rate of shear versus shear stress (See Chapter 10). Gases and ordinary hydrocarbon liquids are Newtonian in nature. Correlations for estimating the viscosity and density of fluids are shown in Chapter 3.

Table 13.1 summarizes the values of A , a and b in Equation 13.7 that apply for the *turbulent flow* of fluids in and around cylindrical tubing.^(13.4) The values shown are based on data from heat exchangers in general chemical service. However, they can prove useful for estimating the performance of a specific heat exchanger. One very important use is prediction of the change in performance of a heat exchanger with change in fluid or flow conditions.

The following conversion factors are useful in solving Equation 13.7.

$$\begin{aligned} 1 \text{ centipoise (cp)} &= 2.419 \text{ lbm/(ft-h)} = 0.001 \text{ kg/m-s} \\ 1 \text{ m} &= 3.281 \text{ ft, } 1 \text{ ft} = 0.305 \text{ m} \\ 1 \text{ kg/m}^3 &= 0.0624 \text{ lbm/ft}^3, 1 \text{ lbm/ft}^3 = 16.02 \text{ kg/m}^3 \\ 1 \text{ Btu/(lbm-}^\circ\text{F)} &= 4.19 \text{ kJ/(kg-K)} \end{aligned}$$

Fluid Properties

Prediction of the density and viscosity terms is possible from correlations like those shown in Chapter 3. Many references contain thermal conductivity data for gases and liquids.^(13.5, 13.6) Shown following are some representative thermal conductivities for gases and liquids at atmospheric pressure and about 38°C [200°F].

The "k" of liquids does not vary much with pressure. There is a change with temperature but it is not large over the usual temperature range of fluid-fluid exchangers. An equation for estimating the "k" of crude oil and condensates is:

$$\begin{aligned} \text{Metric: } k &= 0.131 - (0.000142)(^\circ\text{C}) \\ \text{English: } k &= 0.0773 - (0.0000456)(^\circ\text{F}) \end{aligned} \quad (13.9)$$

Where: k = W/(m-K) or Btu/(h-ft-°F)

Data on the "k" values of gases at high pressure are sparse. From the kinetic theory of gases "k" is theoretically proportional to gas viscosity. One can correct the "k" for pressure using corresponding viscosity correction as an approximation.

The Prandtl number ($C_p\mu/k$) is essentially constant with pressure and temperature. For natural gas a value of 0.79 is suitable for this number.

TABLE 13.1
Coefficients for Use in Equation 13.7 for Turbulent Flow in
Tube Type Heat Exchangers (Fluid-Fluid)

Service	Coefficient A	Exponent a	Exponent b
Inside Vertical Tubes			
Cooling gas	0.0265	0.8	0.3
Cooling liquid (high visc.)	0.027	0.8	0.33
Heating gas	0.0243	0.8	0.4
Heating liquid (low visc.)	0.023	0.8	0.4
Heating liquid (high visc.)	0.027	0.8	0.33
Boiling liquid	0.029	0.8	0.4
Inside Horizontal Tubes			
Cooling gas	0.023	0.8	0.3
Cooling liquid (low visc.)	0.0265	0.8	0.3
Cooling liquid (high visc.)	0.027	0.8	0.33
Heating gas	0.0225	0.8	0.4
Heating liquid (low visc.)	0.0255	0.8	0.4
Heating liquid (high visc.)	0.027	0.8	0.33
Flow normal to a single tube, cooling or heating			
Gas	0.26	0.6	0.3
Liquid	0.38	0.56	0.3
Gas or liquid flow normal to banks of tubes			
Staggered	0.33	0.6	0.33
In-Line	0.26	0.6	0.33

Material	Thermal Conductivity	
	W/(m·K)	Btu/(h·ft·°F)
Liquid		
Freon 12	0.083	0.048
C ₃ -C ₁₀ (n-paraffins)	0.14-0.15	0.079-0.083
Water	0.62	0.36
Ammonia	0.50	0.29
Benzene	0.16	0.091
Crude oil	0.13	0.076
Ethylene Glycol	0.26	0.15
Diethylene Glycol	0.21	0.12
Triethylene Glycol	0.19	0.11
Monoethanolamine	0.24	0.14
Diethanolamine	0.22	0.13
Gas		
Air	0.026	0.015
Ammonia	0.028	0.016
Benzene	0.012	0.0069
Butane	0.017	0.010
Carbon Dioxide	0.019	0.011
Methane	0.035	0.020
Propane	0.021	0.012
Water Vapor	0.021	0.012
Freon	0.011	0.0064

Effect of Velocity

The velocity of the fluid, inside or outside the tubes, has a significant effect on exchanger performance. For flow inside tubes,

$$h \propto v^{0.8}$$

Within reasonable limits, as velocity increases the h increases. The reason? The film is thinner and there is less resistance to heat flow.

But, as velocity increases so does pressure drop. For flow inside tubes,

$$\Delta P \propto v^{1.8}$$

If pressure drop is critical and must be recovered by pumping or compression, an economic compromise is necessary. So ... specification of the heat exchanger must consider pressure drop considerations.

One must also consider varying throughput. If the exchanger is too large at the lowest flow rates expected, you can expect " h " to be lower than at rated capacity. For developing fields it might be more efficient to plan on two or more exchangers in parallel. The manifold is constructed initially with exchangers being added as needed to maintain efficient heat transfer. If block valves are added initially, no shutdown is needed for future additions. Sometimes some of the tubes are plugged during early development to maintain desirable velocity.

When using parallel equipment of any kind, positive controls must be added to insure proper flow distribution. You cannot rely on header design alone.

There is a special consideration in cases where a corrosion or hydrate inhibitor is being injected. In order to achieve good distribution in all tubes a certain minimum tube velocity must be maintained. From test, it has been established that

$$(\rho)(v_m)^2 = A \quad (13.10)$$

Where: ρ = fluid density
 v_m = min. tube velocity
 A = correlation constant

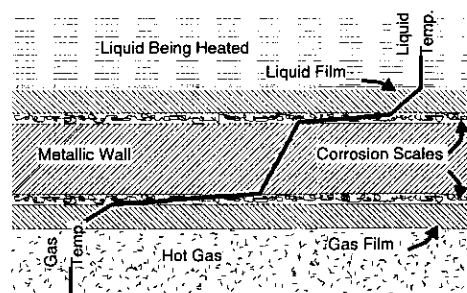
Metric	English
kg/m^3	lb/ft^3
m/s	ft/s
670	450

For gases, ρ is found from methods comparable to those discussed in Chapter 3, at average exchanger pressure and temperature.

Fouling Factors

The fouling factor (F_f) shown in Equations 13.3 and 13.4 must be estimated from experience. The figure at right shows a temperature gradient across a wall, including the possible corrosion or depositional scales. If scale forms, how fast will it form? This is a question that requires a detailed analysis.

Some erroneously use a fouling factor as an arbitrary safety factor. Use of a fouling factor is all right so long as the number used is a realistic one compatible with expected performance. If too large a number is used it controls " U " and invalidates the calculation.



In Equations 13.3 and 13.4 the units of F_f are the reciprocal of those for "U" or "h." It is customary to talk about a fouling factor by quoting a whole number. The number quoted must be inserted in these equations with two zeros in front of it. For example, a fouling factor of 5 would be written as 0.005.

We hesitate to quote any fouling factor for fear it will be misused. The ones shown below are ones that we often note. We offer them without comment.

Units of "h"	F_f
Btu/hr-ft ² -°F	0.001-0.0015
W/(m ² ·°C)	0.006-0.009

These numbers are used often for nonscale forming liquids, free of suspended solids.

SHELL-AND-TUBE EXCHANGERS

Figures 13.3-13.6 show the basic characteristics of shell-and-tube exchangers. The major manufacturers of such equipment have a trade association (TEMA) which has a set of standards. They are not a code but are used commonly in bid specifications. Class R exchangers are used most commonly in the petroleum industry.^(13.7)

Figure 13.3 shows the general head and shell types. The other figures show the details of various types of exchangers using the nomenclature shown at the top of Figure 13.4.

The choice of configuration depends on a number of considerations – fluids involved, corrosion potential, problems of cleaning, pressure drop, heat transfer efficiency. Heat exchanger selection is not routine.

Do you need removable or nonremovable tube bundles? The latter are relatively inexpensive and provide maximum protection against shell-side leakage but they are not accessible for mechanical shell-side cleaning. A type of expansion joint is sometimes needed to relieve differential thermal expansion stresses.

Removable tube bundles consist of U tubes (hairpin type) or straight tubes with a floating head. The former is the least expensive, can be used with very high pressures on the tube-side and no shell-side impingement plates are necessary. But, mechanical cleaning is difficult and it is very difficult to replace tubes.

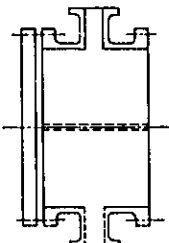
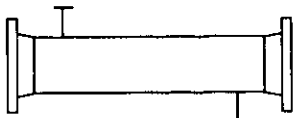
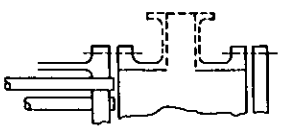
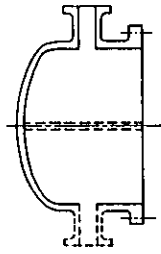
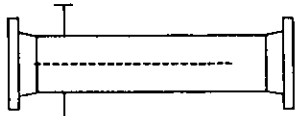
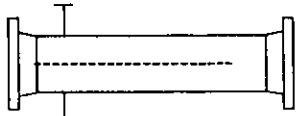
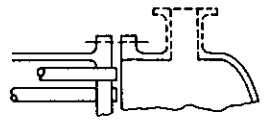
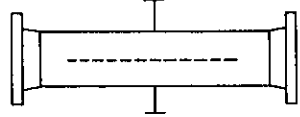
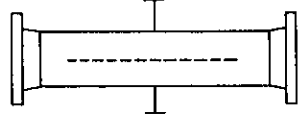
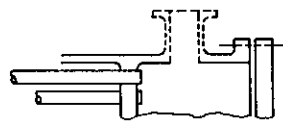
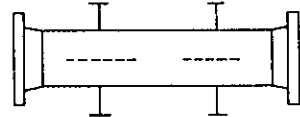
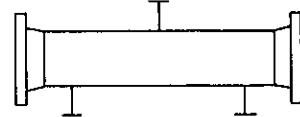
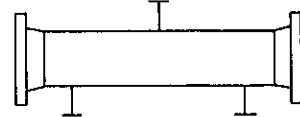
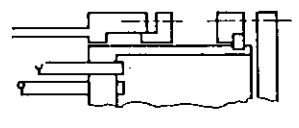
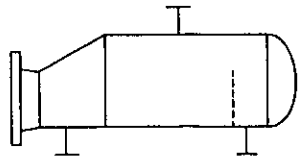
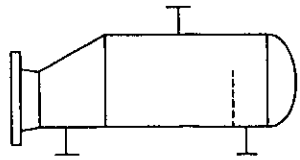
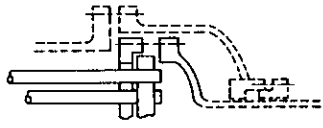
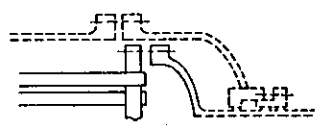
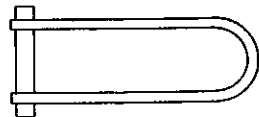
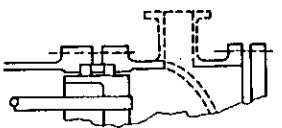
The floating head exchanger is the most versatile and most expensive. Obtaining a positive seal between tube-side and shell-side fluids is critical in many cases. The pressure differential between shell- and tube-sides is limited by the seal.

Although there are exceptions, most tubes used are 1.5-2.5 cm [5/8-1.0 in.] diameter, 1.9 cm [3/4 in.] is the most common. The larger size normally is used when fouling is anticipated, to facilitate mechanical cleaning. The tube length may be as large as 12 m [40 ft] but tubes about half this length are more commonly employed.

The tube bundle can be arrayed in a triangular, square or rotated-square layout. Triangular usually gives better shell-side "h" values and more heat transfer area for a given shell diameter. However, the other arrangements are easier to clean and have a lower pressure drop.

LMTD Correction Factor

The factor "F" in Equation 13.5 is essentially unity in a pipe-in-pipe exchanger and in counterflow shell-and-tube exchangers with an equal number of shell and tube passes. If the number of shell and tube passes differ or if the exchanger is a crossflow type (e.g., TEMA "J," aerial coolers), F will have a value less than 1.0, depending on the flow configuration used. As a practical economic consideration it is seldom that a configuration should be chosen where F is less than about 0.80.

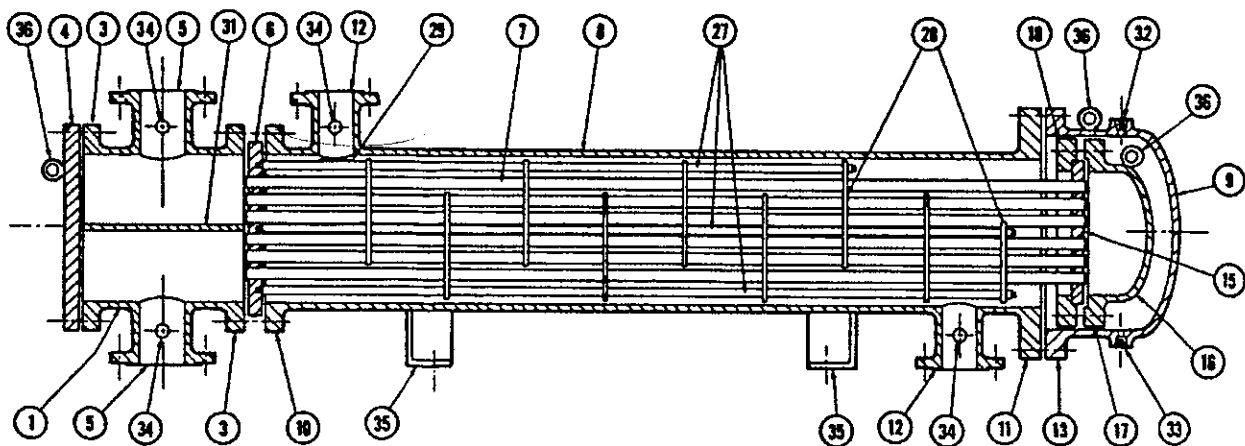
FRONT END STATIONARY HEAD TYPES		SHELL TYPES		REAR END HEAD TYPES	
A		E		L	
	CHANNEL AND REMOVABLE COVER				
					
	BONNET (INTEGRAL COVER)				
B		F		M	
C		G		N	
	REMOVABLE TUBE BUNDLE ONLY				
					
	FIXED TUBESHEET ONLY				
D		J		P	
	CHANNEL INTEGRAL WITH TUBE- SHEET AND REMOVABLE COVER				
E		K		S	
	SPECIAL HIGH PRESSURE CLOSURE				
F				T	
G				U	
H				W	

Courtesy Tubular Exchanger Mfgs. Assn. (TEMA)

Figure 13.3 Basic Mechanical TEMA Characteristics

Number Guide For Use With Figures 13.4, 13.5, and 13.6

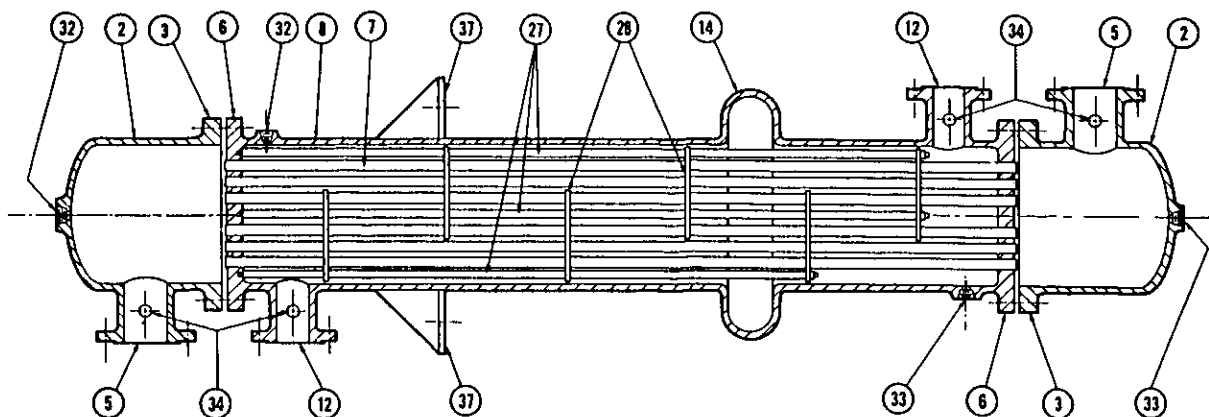
1. Stationary Head-Channel	21. Floating Head Cover-External
2. Stationary Head-Bonnet	22. Floating Tubesheet Skirt
3. Stationary Head Flange-Channel or Bonnet	23. Packing Box Flange
4. Channel Cover	24. Packing
5. Stationary Head Nozzle	25. Packing Follower Ring
6. Stationary Tubesheet	26. Lantern Ring
7. Tubes	27. Tie Rods and Spacers
8. Shell	28. Transverse Baffles or Support Plates
9. Shell Cover	29. Impingement Baffle
10. Shell Flange-Stationary Head End	30. Longitudinal Baffle
11. Shell Flange-Rear Head End	31. Pass Partition
12. Shell Nozzle	32. Vent Connection
13. Shell Cover Flange	33. Drain Connection
14. Expansion Joint	34. Instrument Connection
15. Floating Tubesheet	35. Support Saddle
16. Floating Head Cover	36. Lifting Lug
17. Floating Head Flange	37. Support Bracket
18. Floating Head Backing Device	38. Weir
19. Split Shear Ring	39. Liquid Level Connection
20. Slip-on Backing Flange	



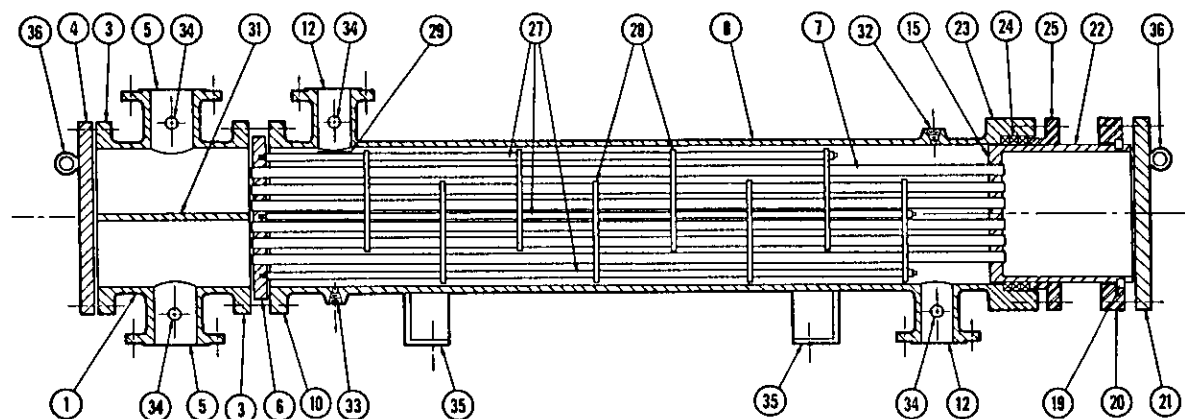
1-Pass Shell, 2-Pass Tube Exchanger

Courtesy Tubular Exchanger Mfgs. Assn. (TEMA)

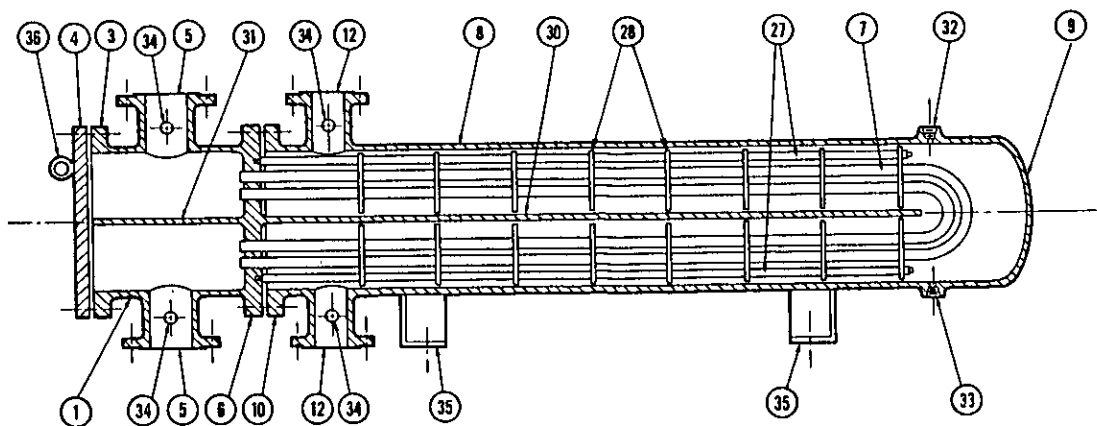
Figure 13.4 One Type of Tubular Exchanger



1-Pass Shell and Tube with Expansion Joint on Shell Side



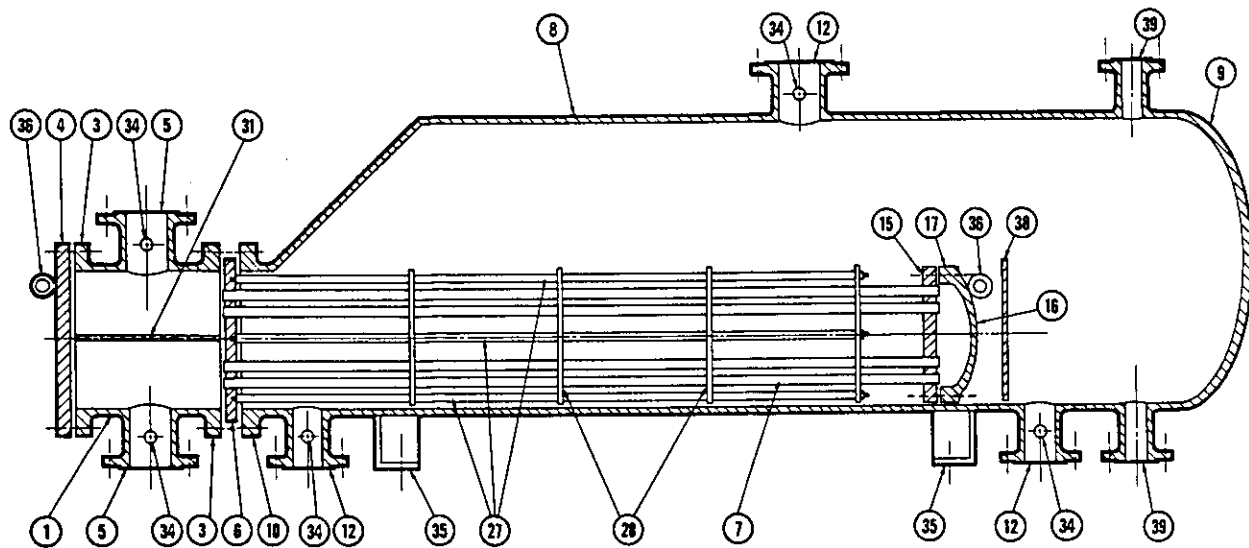
1-Pass Shell and 2-Pass Tube



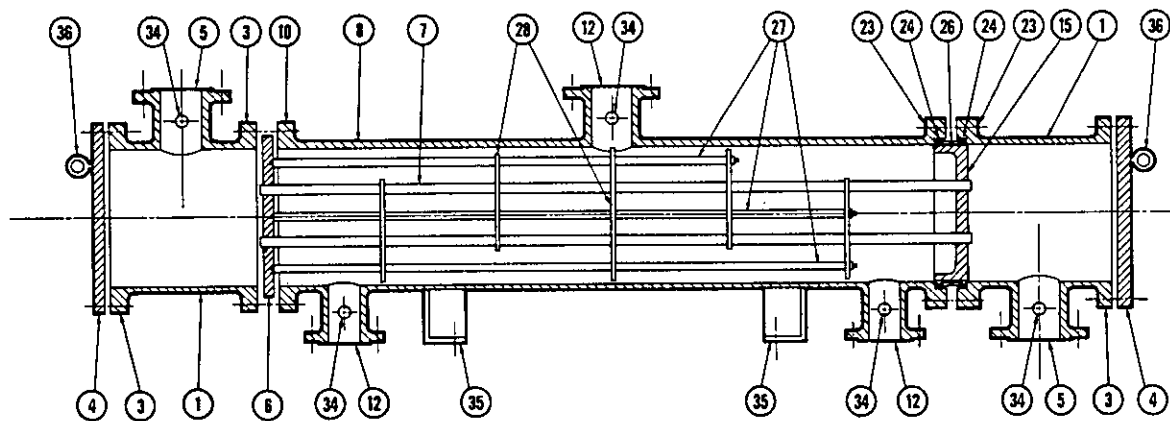
Hairpin

Courtesy Tubular Exchanger Mfgs. Assn. (TEMA)

Figure 13.5 Three Other Examples of Tubular Exchangers



Kettle Reboiler



Thermosiphon Reboiler

Courtesy Tubular Exchanger Mfgs. Assn. (TEMA)

Figure 13.6 Two Common Types of Reboilers

Figures 13.7-13.12 provide a means to estimate the factor F shown on the left ordinate. Values of P and R on these figures are found by the equations

$$P = \frac{t_2 - t_1}{T_1 - t_1}, \quad R = \frac{T_1 - T_2}{t_2 - t_1}$$

For a given value of P and R , find the corresponding value of F . If the values of P and R do not intersect within the grid, simply record F as less than 0.5.

As shown on each figure, T_1 and T_2 represent the shell-side fluid and t_1 and t_2 the tube-side fluid.

This correlation is a modified version that has been in use for a long time.^(13.8) An alternate calculation to that shown has been developed.^(13.9) The net results are essentially the same.

Figures 13.11 and 13.12 are suitable for an aerial cooler or for any case where one fluid flows normal to a bank of tubes.

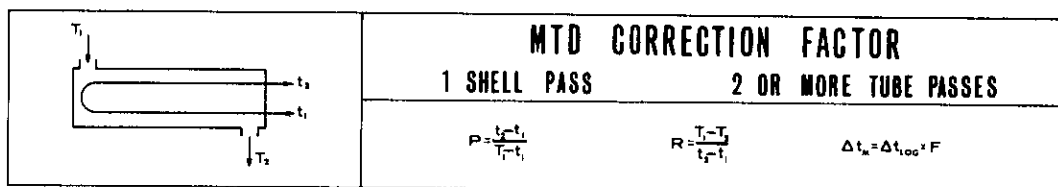
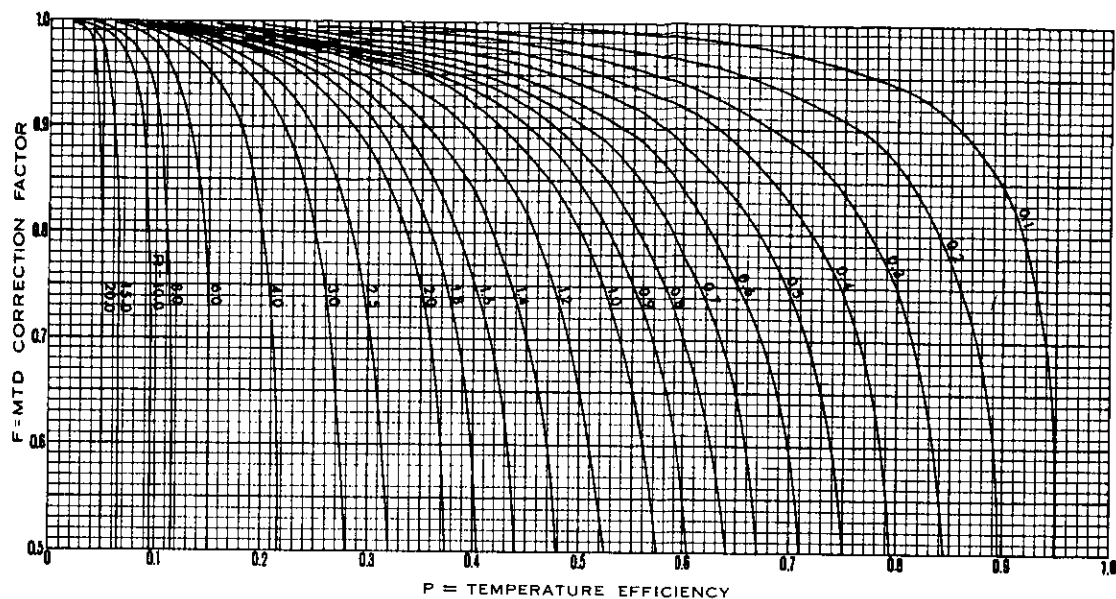


Figure 13.7

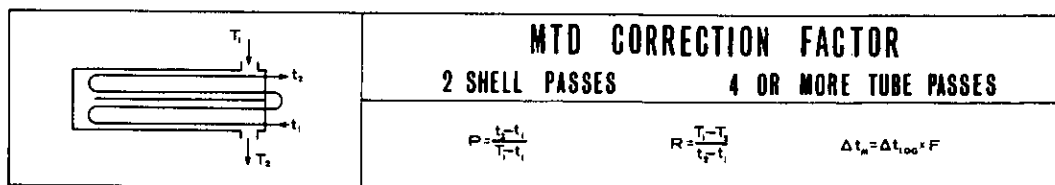
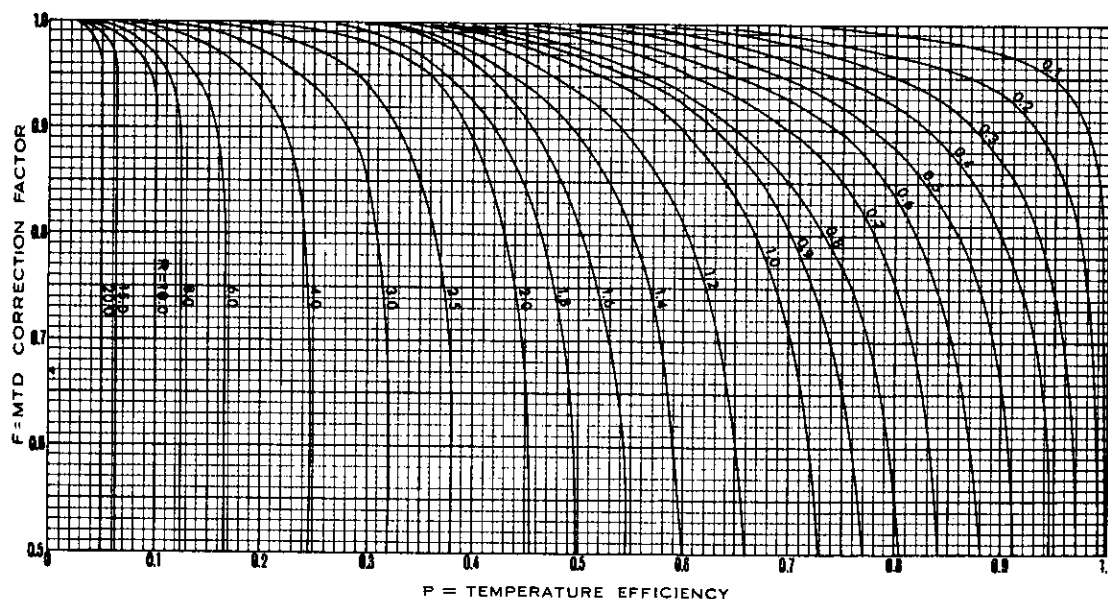


Figure 13.8

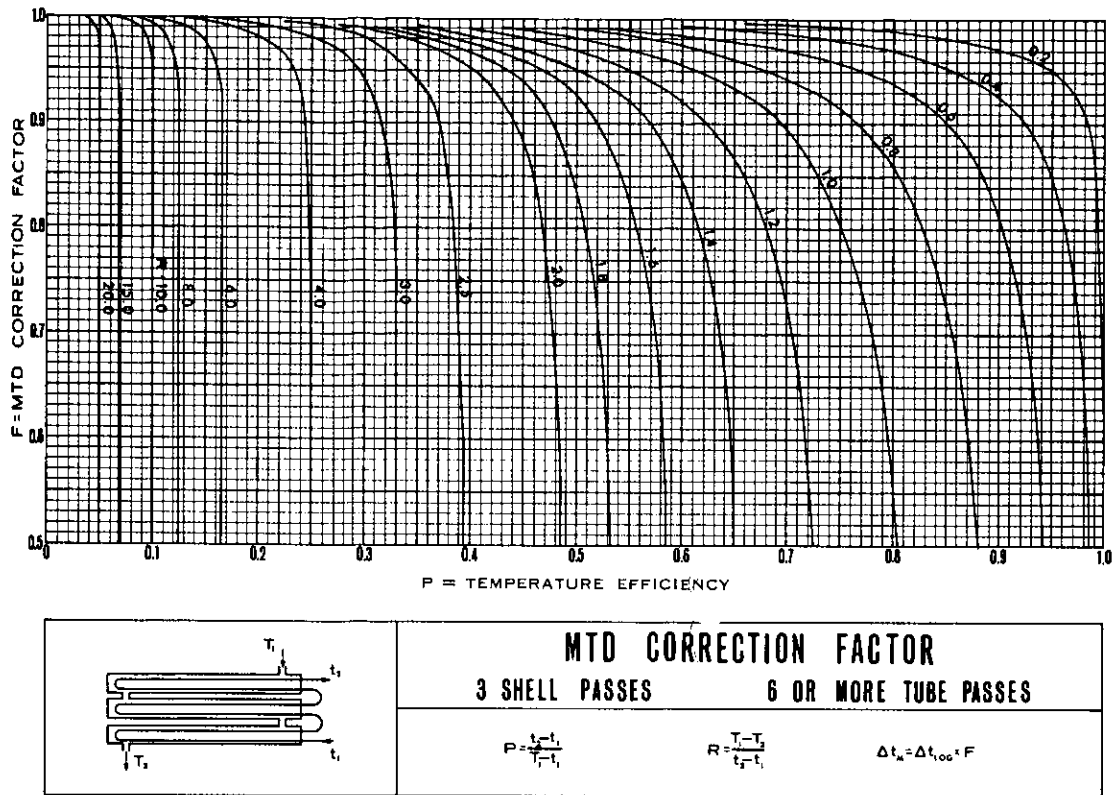


Figure 13.9

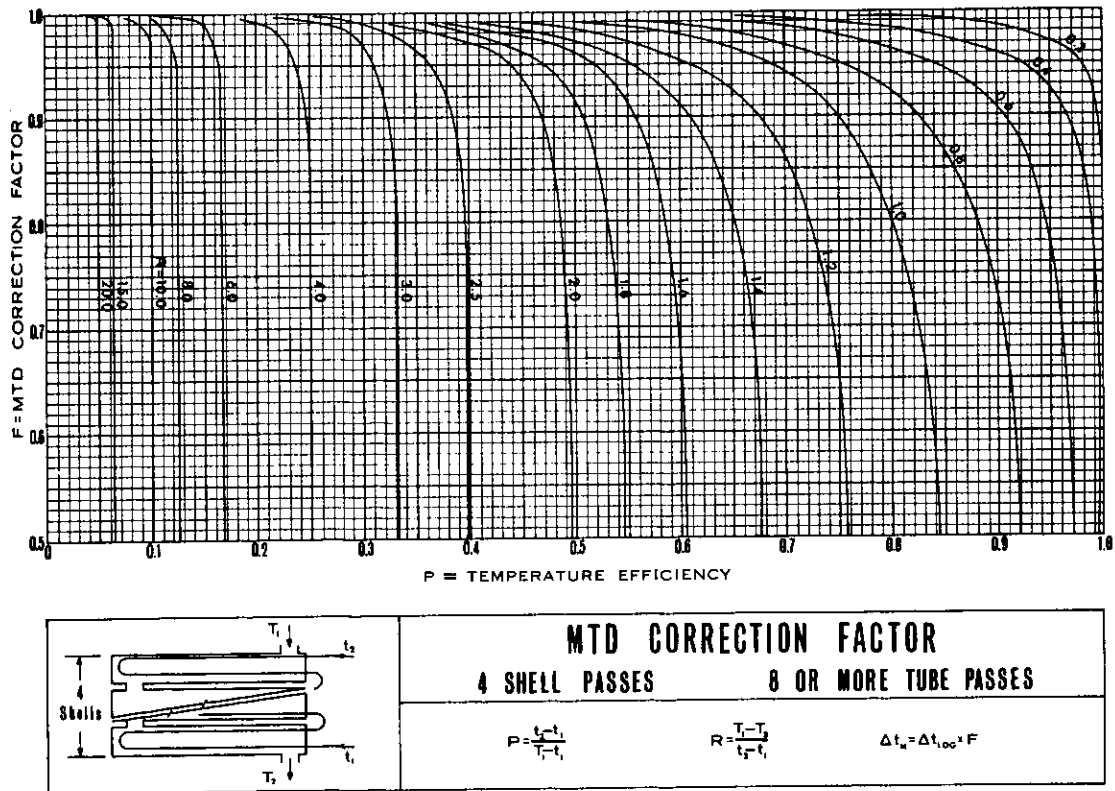


Figure 13.10

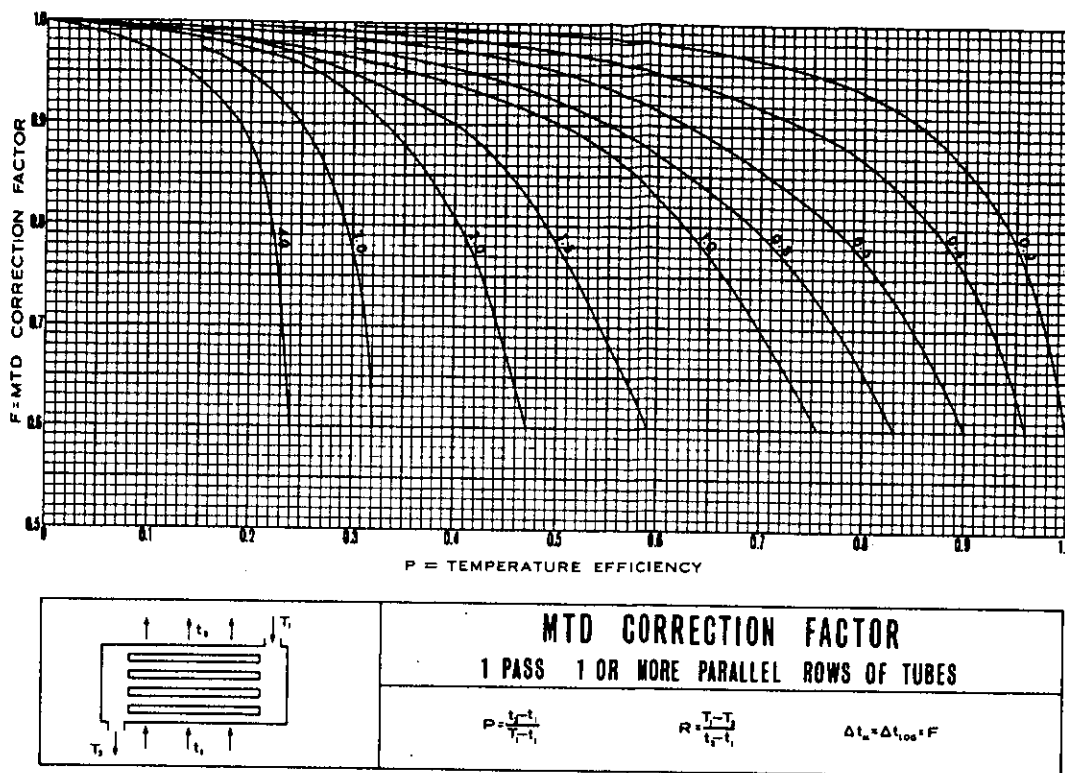


Figure 13.11

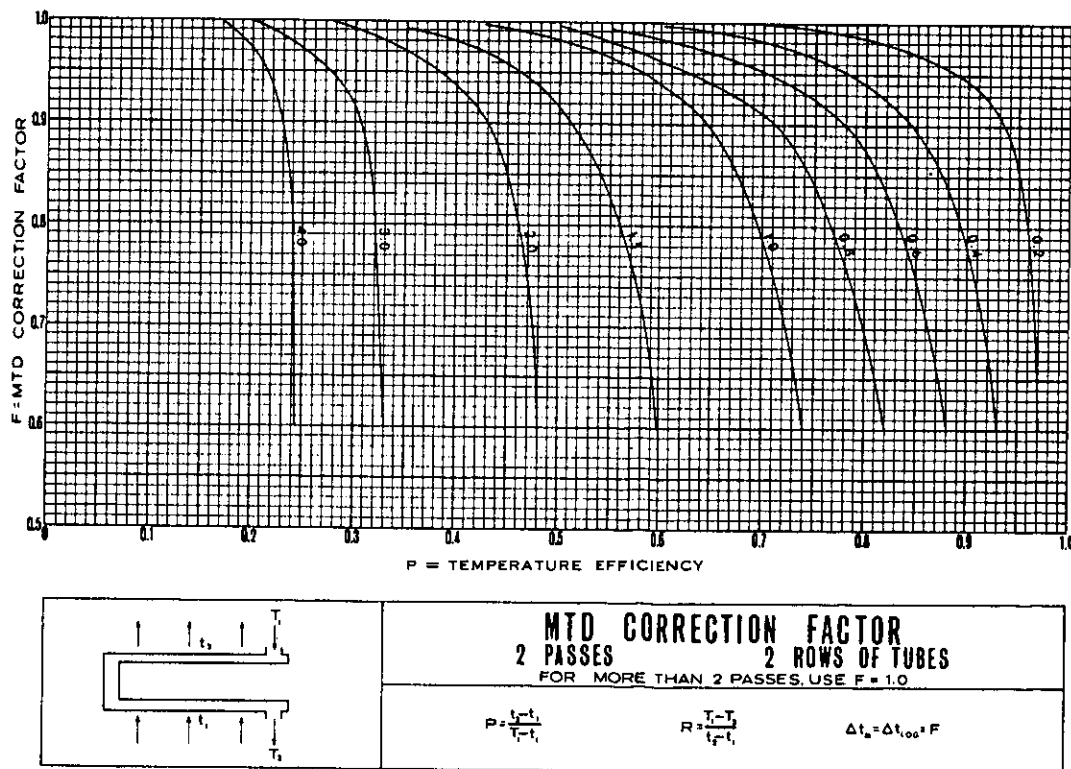


Figure 13.12

Example 13.3: What configuration of shell-and-tube heat exchanger would be suitable for use in the previous lean oil example? (Figures 13.11-13.12 really do not apply)

$$P = \frac{166 - 43}{210 - 43} = 0.74$$

$$R = \frac{210 - 69}{166 - 43} = 1.15$$

From Figure 13.7, $F = < 0.5$

Figure 13.9, $F = 0.6$

Figure 13.8, $F = < 0.5$

Figure 13.10, $F = 0.85$

Only a four-shell-pass, eight-or more-tube-pass exchanger has a large enough value of F . But, would one want this kind of configuration in this service? The answer would depend on unspecified factors. A shell-and-tube exchanger might not be at all suitable.

Fluid Placement

This obviously affects the value of F in the LMTD calculation. But, the major consideration may be the character of the fluid itself. The following general guidelines are useful.

A. Shell-Side

1. Viscous fluid to increase (generally) the value of "U"
2. Fluid having the lowest flow rate
3. Condensing or boiling fluid

B. Tube-Side

1. Toxic and lethal fluids to minimize leakage
2. Corrosive fluids
3. Fouling fluids; increased velocity minimizes fouling but enhances erosion
4. High temperature fluids requiring alloy materials
5. High pressure fluids to minimize cost
6. Fluid on which pressure drop is most critical

These are not mutually exclusive considerations. Some priorities must be established; some compromises are necessary. For example, condensing may be done on the tube side when special metallurgy is required. In this case, vertical tubes normally are a better choice than horizontal tubes.

In some cases a series of exchangers (train) is required. One then must divide the total heat transfer duty to optimize the number and size of each unit. ^(13.10, 13.11)

Factors like these governing testing, troubleshooting, etc. are covered in References 13.12-13.18.

Estimation of Mechanical Design

As part of the early planning function, it may be desirable to estimate the physical size of the exchanger being considered. Figure 13.13 provides an easy method to accomplish this. ^(13.11) The equation for use with this figure is

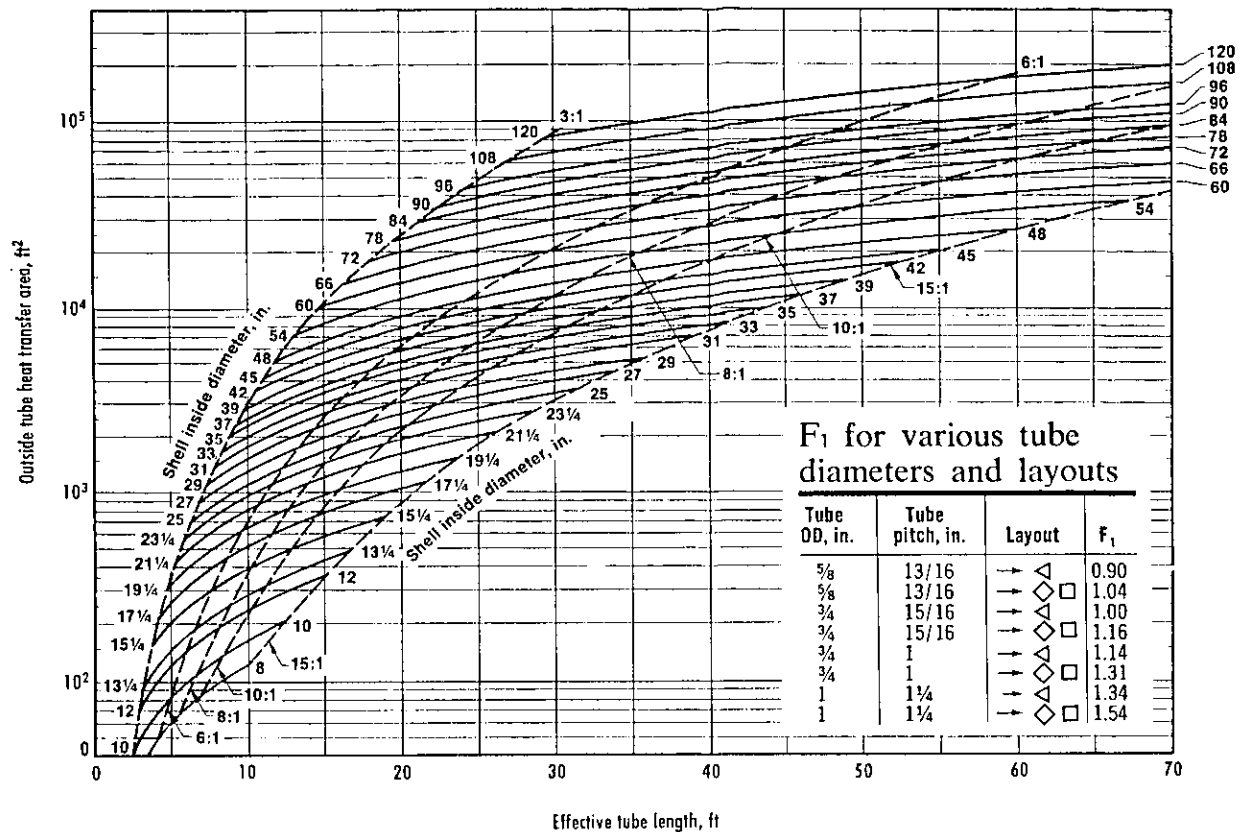
$$A = A_0 F_1 F_2 F_3 \quad (13.11)$$

Where: A = area on left-hand ordinate of figure

A_0 = area calculated from heat transfer equation

F_1, F_2, F_3 = correction factors shown in Fig. 13.13

F_1, F_2 and F_3 are equal to unity for 3/4-in. tubes on a 15/16-in. triangular pitch, one tube pass and a fixed tube sheet exchanger, respectively.



F ₂ for Various Numbers of Tube-Side Passes*				
Inside shell diameter, in.	Number of tube-side passes			
	2	4	6	8
Up to 12	1.20	1.40	1.80	—
13 1/4 to 17 1/4	1.06	1.18	1.25	1.50
19 1/4 to 23 1/4	1.04	1.14	1.19	1.35
25 to 33	1.03	1.12	1.16	1.20
35 to 45	1.02	1.08	1.12	1.16
48 to 60	1.02	1.05	1.08	1.12

*Since U-tube bundles must always have at least two passes, use of this table is essential for U-tube bundle estimation. Most floating-head bundles also require an even number of passes.

F ₃ for Various Tube-Bundle Constructions					
Type of tube bundle construction	Inside shell diameter, in.				
	Up to 12	13 1/4 to 21 1/2	23 1/4 to 35	37 to 48	Above 48
Split backing ring (TEMA S)	1.30	1.15	1.09	1.06	1.04
Outside packed floating head (TEMA P)	1.30	1.15	1.09	1.06	1.04
U-Tube* (TEMA U)	1.12	1.08	1.03	1.01	1.01
Pull-through floating head (TEMA T)	—	1.40	1.25	1.18	1.15

*Since U-tube bundles must always have at least two tube side passes, it is essential to also use the above table for this configuration.

Figure 13.13 Estimation of Tubular Exchanger Size

For a given value of A , various diameter/length combinations are suitable. These would need to be checked for fluid velocities. L/D ratios less than 3:1 may suffer from poor fluid distribution. Ratios in range 6:1-10:1 generally are a good compromise. These L/D ratios are shown as dashed lines in Figure 13.13.

PLATE AND FRAME EXCHANGERS

Where applicable, the plate exchanger has become a viable alternative for heat exchange. In many services it is lighter, more compact, less expensive, and offers better overall performance than more traditional types. It is very competitive in many services in frontier and offshore applications. Offshore, the use of plate exchangers for sea water cooling has become almost universal. References 13.19-23 provide useful additional information and are the source of illustrations in this section.

Figure 13.14 summarizes some basic characteristics of this exchanger. Part (a) shows the basic construction. A series of corrugated, pressed metal plates is clamped together and held in place by bolts through the end plates and a pressure plate. Each such plate is gasketed to prevent leakage. There are four flow channels on each plate which can be blanked off or combined in different ways to form different flow patterns. An assembled plate exchanger is shown in (b) of Figure 13.14.

Shown in (c) are plates with chevron-type grooves. A low chevron angle plate at left is called a high theta (θ) plate; a high angle is a low theta plate. Different manufacturers use different corrugated or embossed patterns for competitive advantage (hopefully).

Also shown in Part (c) of Figure 13.14 is the fact that plates may be mixed and matched to provide more efficient heat transfer. One of the "tricks" of the trade is providing the best combination. Remember, the heat transfer plates are grooved on both sides. Since the plates are symmetrical, the direction of the grooves and their angle can be reversed on alternate plates.

The term theta is defined by Equation 13.12 in a later section. It is a measure of heat transfer effectiveness. As in all heat exchange, effectiveness is purchased with pressure drop. A high theta plate has a higher pressure drop than a low theta plate.

Part (d) is an example of mixing plates for cooling a large amount of process water with cooling water.^(13.19) The optimum mixture of plate types reduced the number of plates (and thus cost) significantly.

The basic idea is to form a series of interlocking flow channels which produce high velocity. The metal-to-metal contact also helps plate rigidity. Any metal that can be cold worked can be used. Titanium is a common plate material for sea water cooling. Stainless steels, Monel, nickel, Incoloy, etc., also are used in some services.

The size and thickness of plates depend on the metal used, system pressure, and process specifications. Pressure deflection must be limited to prevent leaking.

Plate thickness ranges from 0.5-3.0 mm with an average gap between plates of 1.5-5.0 mm. Most plates have an area less than 1.5 m². The pressure limitation is obvious. A pressure of about 2.0 MPa [300 psia] is a good maximum although slightly higher pressures have been used. Where possible, a maximum pressure of about 1.0 MPa [150 psia] is preferred.

Temperature is limited by metallurgy and gasket materials. Installations are operating up to about 250°C [480°F]. But, the utility of the plate exchanger diminishes rapidly above 150°C [300°F].

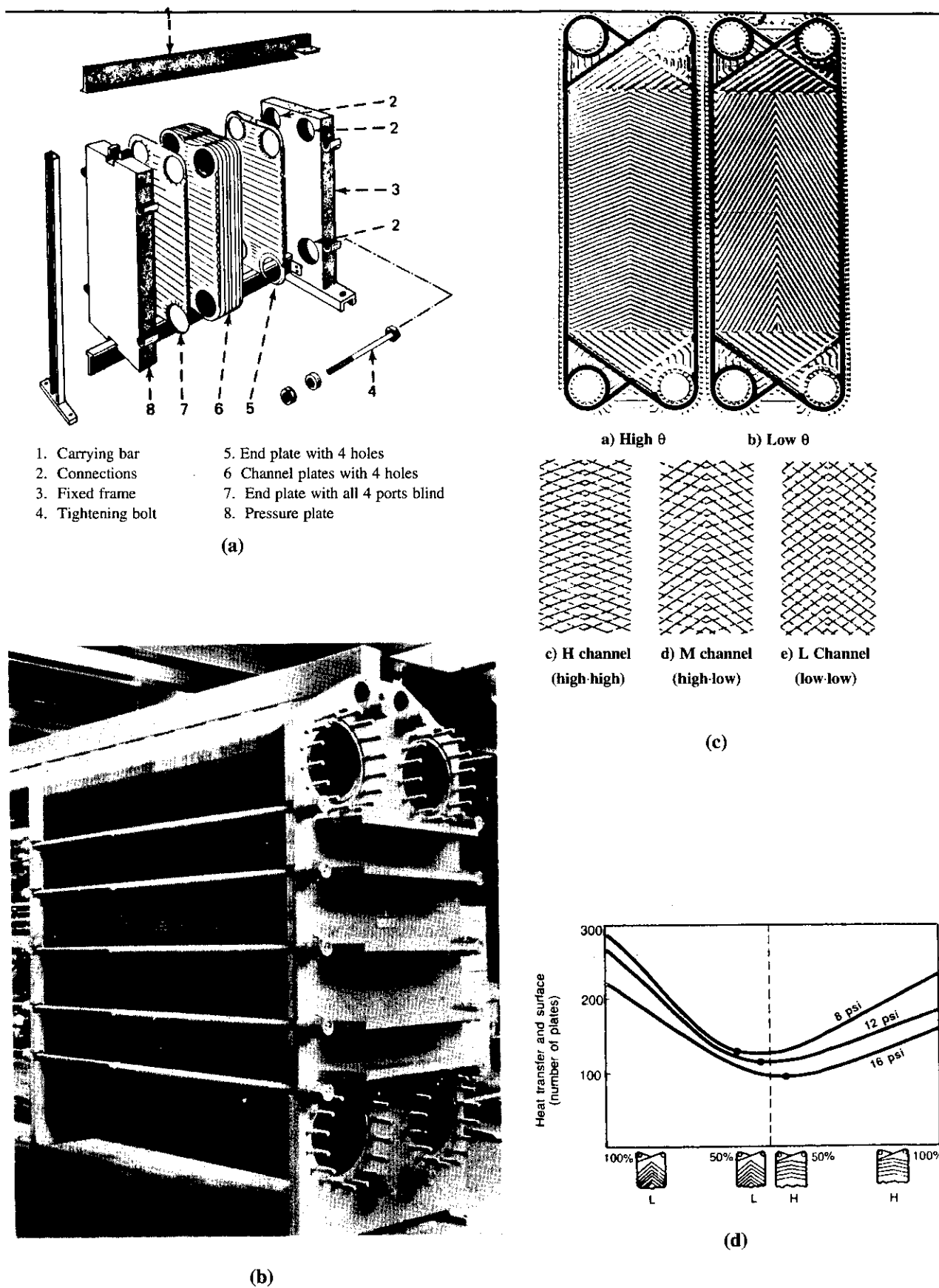


Figure 13.14 General Characteristics of Plate Exchangers

Theta Factor

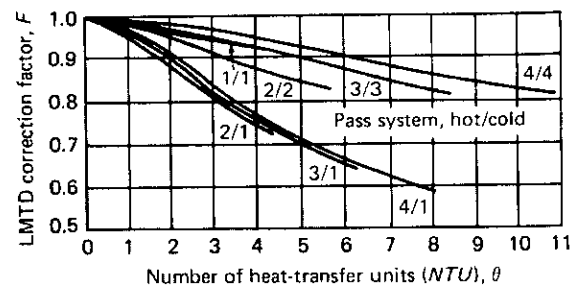
This factor also is called NTU (number of transfer units). This theta can be stated in terms of the required duty or the enthalpy change of one or both fluids. A consistent set of units must be used to make theta a dimensionless number. The total area is twice the area of the thermal plates. The end plates are not involved in heat transfer.

$$\theta = \frac{t_i - t_o}{\Delta t_m} = \frac{U A}{m C_p} \quad (13.12)$$

Where: t_i = inlet temperature to channel of fluid
 t_o = outlet temperature to channel of fluid
 U = overall heat transfer coefficient
 A = total area of thermal plates
 m = mass flow rate of fluid per hour
 C_p = specific heat of fluid

Determination of LTD

LMTD is determined in the same manner as before, the only difference is in "F." The figure at right is an approximate correlation for F in plate exchangers. For an equal number of passes on the hot and cold fluids and for a low NTU (or θ), F approaches one. An unequal number of passes usually occurs only when the two fluids involved have widely different flow rates. Where a different number of passes is required, the plate exchanger may not be the best choice.



Heat Transfer Coefficients

The film coefficients use the same form as Equation 13.7. They may be estimated by the equations:

For turbulent flow ^(13.22)
$$h = A \frac{k}{d_e} \left(\frac{d_e w}{\mu} \right)^{0.65} \left(\frac{C_p \mu}{k} \right)^{0.4} \quad (13.13)$$

For laminar flow ^(13.24)
$$h = A' C_p w \left(\frac{d_e w}{\mu} \right)^{-0.62} \left(\frac{C_p \mu}{k} \right)^{-0.67} \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (13.14)$$

Where: k = thermal conductivity
 d_e = $(4Wb)/(2W + 2b)$
 W = plate width
 w = mass flow rate
 μ = fluid viscosity
 C_p = specific heat
 μ_w = fluid viscosity at wall
 A = proportionality constant
 A' = proportionality constant
 h = film coefficient

Metric	English
W/(m·K)	Btu/h-ft-°F
m	ft
m	ft
kg/(s·m ²)	lbm/(h-ft ²)
kg/(m·s)	lbm/(ft-h)
kJ/(kg·K)	Btu/(lbm-°F)
kg/(m·s)	lbm/(ft-h)
0.2536	0.2536
0.742	0.742
W/(m ² ·K)	Btu/(h-ft ² -°F)

The overall heat transfer coefficient is

$$U = \frac{1}{(1/h_1) + (L/k_w) + (1/h_2) + F_f} \quad (13.15)$$

which is the same form as Equations 13.3 and 13.4. For a flat plate all areas are equal and cancel out.

One can calculate area using Equation 13.2. An alternate approach designed primarily for computer usage has been developed by Jackson and Troup.^(13.23) It is based on theta and does not use the LMTD equation. Comparable results should be obtained.

Pressure Drop

The pressure drop depends on plate design, arrangement and flow pattern. Reference to manufacturer's specifications is recommended although Reference 13.25 shows a method for estimation. Pressure drop is about the same as (or less than) that in a comparable duty tubular exchanger.

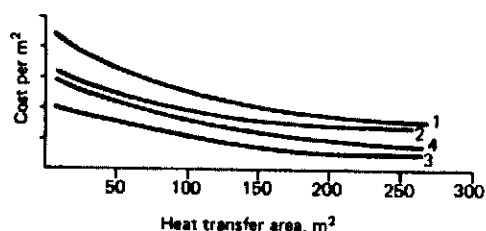
Corrosion/Fouling

Fouling is minimized by high velocity and a clean surface. So, a plate exchanger will have less fouling tendency than a tubular. However, the higher velocity will increase erosion. So, more erosion resistant materials than the cupro-nickel tubular alloys usually are specified.

Since the plates are so thin, corrosion standards are critical. A corrosion rate of over 0.05 mm [2 mils] per year would be unacceptable in most instances. Because of erosion/corrosion considerations, the choice of metallurgy is more critical in plate than in tubular exchangers.

General Considerations

One must compare all factors to choose the proper heat exchanger. Within its range of applicability, the plate exchanger may be less expensive than the tubular, particularly where alloy construction is needed. One comparison is shown below for crude oil to water exchange with a heat load of 3.1 MW [10.7 million Btu/h]. Notice the material upgrade for the plate exchanger. Although it cost twice as much per unit area, it had only about 1/6 the area of the tubular. It ended up being smaller, lighter and less expensive.



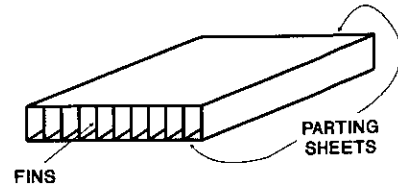
	Shell and tube	Plate heat exchanger
Design pressure	100 psi/250°F	100 psi/250°F
Passes	One shell side/four tube side	One/one
Overall "U"	58 Btu/hr-ft²-°F	336 Btu/hr-ft²-°F
Required surface	3940 ft²	635 ft²
Materials	All carbon steel	Carbon steel frame/316 SS plates
Pressure drops	Hot side: 21 psi Cold side: 5 psi	Hot side: 9.7 psi Cold side: 8.2 psi
Size	3 ft diameter/24 ft length	5-ft length/3-ft width
Weight	23800 lbm	3837 lbm
Plot space required	4 ft x 60 ft	5 ft x 6 ft
Cost/ft²	\$10.60	\$21.42

In spite of possessing many favorable features, the plate exchanger is not an automatic choice. With high pressure gas and/or condensing or boiling fluid, the tubular may be superior even in the P and T range of the plate type.

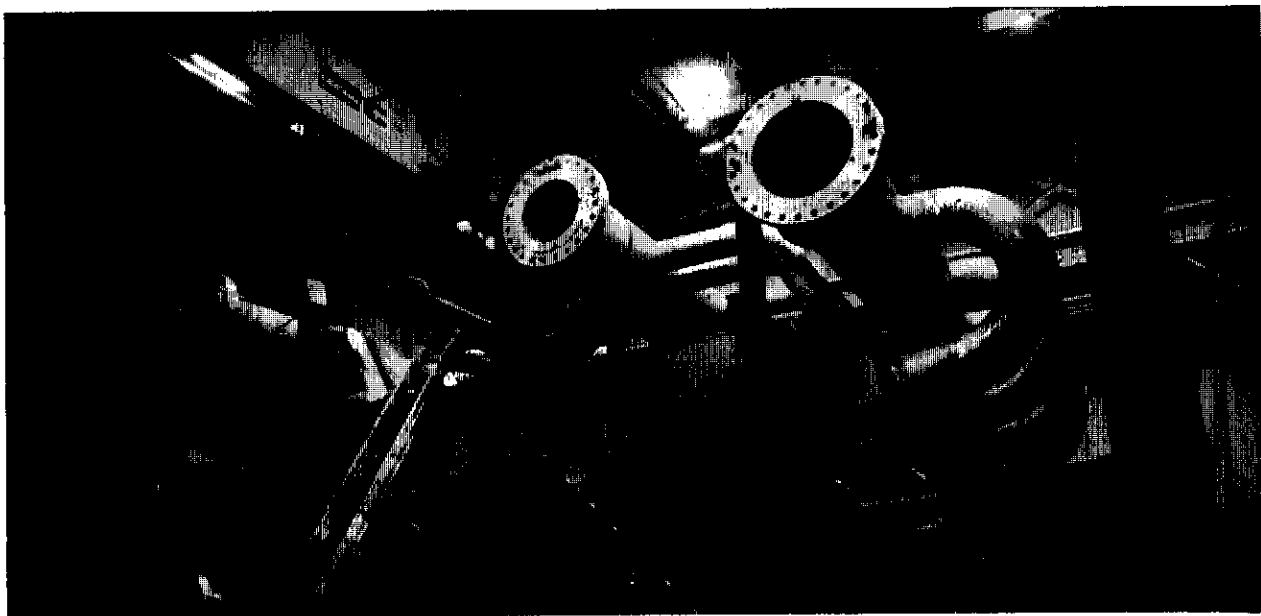
PLATE-FIN EXCHANGERS

Brazed aluminum plate-fin heat exchangers are frequently used in low temperature gas processing service. Often referred to as a "cold-box" they can be designed to handle up to 10 fluids in a single exchanger and can operate at temperatures as low as -269°C [-452°F].

A brazed aluminum heat exchanger is composed of alternating layers of corrugated fins and flat separator sheets called parting sheets. A stack of fins and parting sheets comprise the heat exchanger, sometimes referred to as the "core." Each fluid pass in a core has the appearance of a section of the wall of a cardboard box. The inside and outside panels represent the parting sheets and the corrugations represent the fins. The number of layers, type of fins, stacking arrangement, and stream circuiting will vary depending on the application requirements. A typical plate fin exchanger is shown below.



Fluid configuration may counterflow, crossflow, and cross-counterflow circuiting. Temperature approaches of 1.7°C [3°F] on single-phase fluids and 2.8°C [5°F] on two-phase fluids can be achieved. Typically, corrected mean temperature differences of 2.8 - 5.6°C [5 - 10°F] are employed in brazed aluminum heat exchanger applications.



Brazed aluminum heat exchangers are compact and light-weight. A typical high pressure brazed aluminum heat exchanger with a design pressure of 4100 - 9650 kPa [600 - 1400 psig] will provide 90 - 120 m^2/m^3 [300 - 400 ft^2/ft^3] of heat transfer area per exchanger volume. This is six to eight times the surface density of comparable shell and tube exchangers. Additionally, a typical high pressure brazed aluminum heat exchanger will have a density of 1200 - 1450 kg/m^3 [75 - 90 lbm/ft^3] versus 4000 kg/m^3 [250 lbm/ft^3] for comparable shell and tube exchangers. The net effect of these differences is that a brazed aluminum heat exchanger will provide approximately 25 times more surface per weight of equipment than comparable shell and tube exchangers. This decrease in exchanger weight and volume reduces foundation, support, plot plan, and insulation requirements.

Brazed aluminum heat exchangers are designed and constructed to comply with the "ASME Boiler and Pressure Vessel Code," Section VIII, Division I, or other applicable standards. The aluminum alloys used comply with ASME Section II, Part B, "Nonferrous Materials," or the requirements of the specified code authority.

Aluminum alloy 3003 is generally used for the parting sheets, corrugated fins, and bars which form the rectangular heat exchanger block. These parts are metallurgically bonded by a brazing process at temperatures of about 593°C [1100°F]. The brazing alloy is an aluminum silicon metal and is provided on or with the parting sheets.

Brazed aluminum heat exchangers should be used with clean fluids since they are more susceptible to plugging than other types of heat exchanger equipment; however, proper filters will prevent heat exchanger fouling. Brazed aluminum should not be used with fluids which are corrosive to aluminum. Mercury and caustic soda are extremely corrosive to aluminum and should not be introduced into the exchanger. Hydrogen sulfide and carbon dioxide are not a corrosion problem in streams with water dewpoint temperatures below the cold end temperature of the exchanger.

Design of plate-fin exchangers is not routine, particularly when two-phase streams are present. Cooling and heating curves (T vs. Q) must be developed for each fluid present. The heat exchanger is then broken into "zones" where the cooling and heating curves are essentially linear. The corrected LMTD and heat transfer coefficients are determined for each zone and the zone area is calculated from Equation 13.2. The areas of each "zone" are then summed to give the total exchanger area.

This procedure becomes increasingly complex as more streams are included in the exchanger. Accurate thermodynamic and physical property data is essential. The first and second laws of thermodynamics cannot be violated. The total duties of all fluids must sum to zero and heat can only flow from high temperature to low temperature.

Heat leak in cryogenic heat exchangers is another factor which will affect the cooling curve. It acts as an unwanted heat flow into the heat exchange fluids and will reduce the effective LMTD. For well insulated exchangers, heat leak normally has a negligible effect on the LMTD. However, the amount of heat leak should always be checked and combined as another warm stream on the cooling curve to determine its effect on the LMTD.

PIPE-IN-PIPE EXCHANGERS

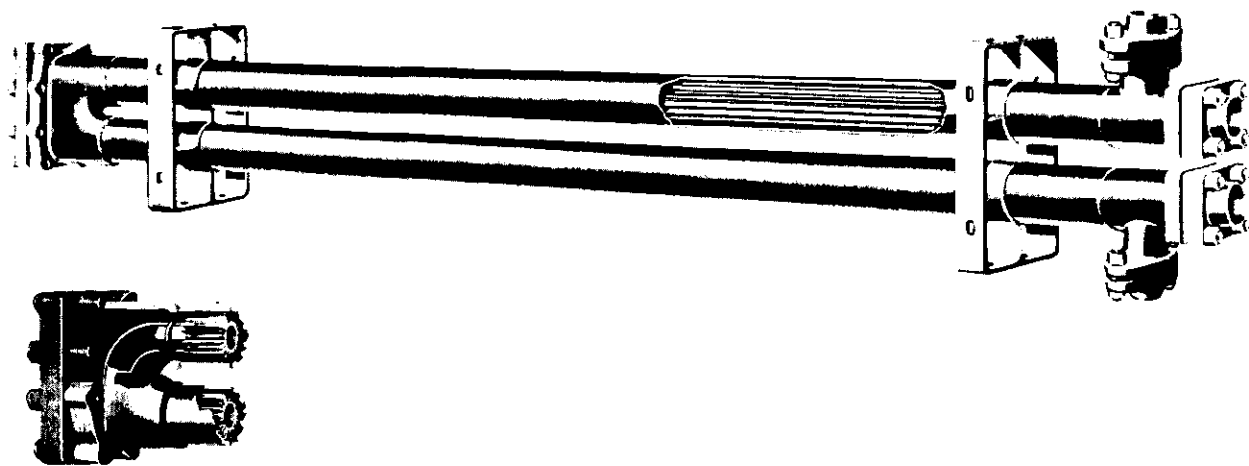
This type of exchanger may be advantageous for relatively low heat loads where one stream is a gas or viscous liquid. Figure 13.15 shows various details of an exchanger employing longitudinal fins.

In this exchanger a piece of pipe serves as a shell. Inside is a single concentric pipe or a group of pipes. A commercial type used commonly consists of a single U-tube unit which may be manifolded in series and parallel to satisfy the heat duty.

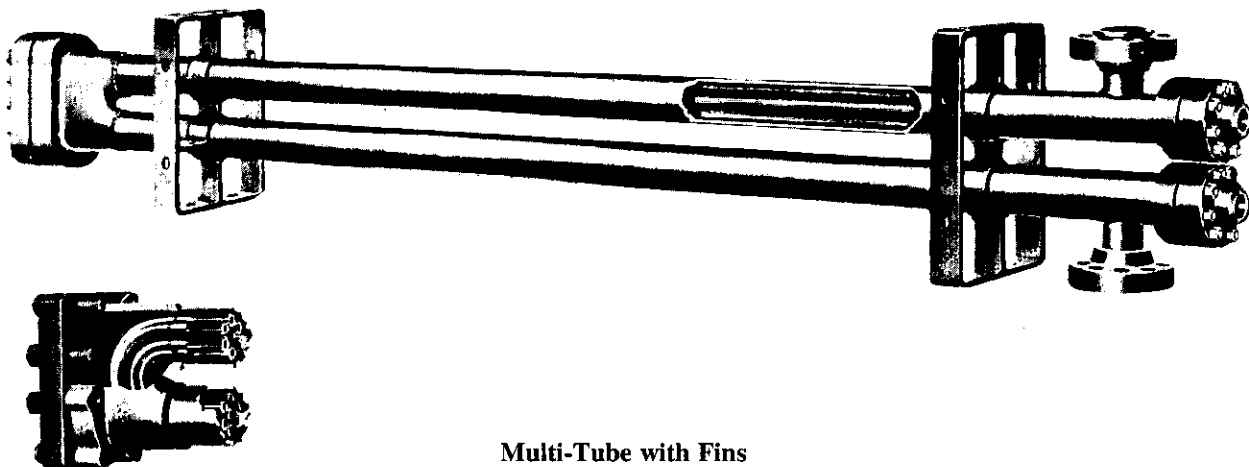
This is very efficient. The cost is competitive so long as the total area needed for heat exchange is not too large. For very large areas, weight and/or volume may prove to be unfeasible for a given application.

Figure 13.15 also shows finned tubes. Those shown are all on the outside of the tubes, but they also can be used inside. The shape and style vary widely.

The purpose of the fin is to increase the surface area to the fluid. The area of the bare pipe plus fins is called the *extended area*. By extending the area with fins the resistance per unit heat transferred is decreased. Thus, fins are used with that fluid whose film coefficient is so low that areas must be increased to give an economical rate of heat transfer.



Single Tube with Fins



Multi-Tube with Fins

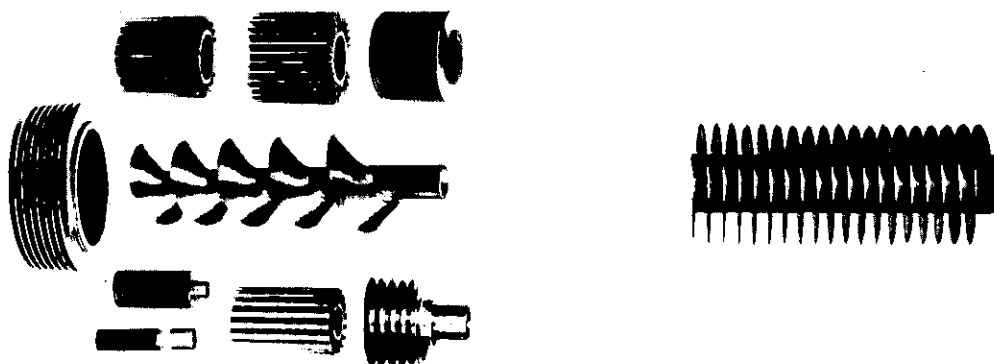


Figure 13.15 Some Details of Finned Tubes and Pipe-in-Pipe Heat Exchangers

Fins are almost always used with aerial coolers because the air film coefficients are low. Finned tubes also may be used in all other types of heat exchangers. They should not be used arbitrarily, however. Fins increase pressure drop and are difficult to clean of scale and solids which become imbedded in them. With corrosive fluids, erosion-corrosion may be enhanced due to impingement and turbulence problems. Unless the fins are firmly attached, vibration may cause mechanical separation of the fin from the tube.

The design of these units is similar to other exchangers. Film coefficients are calculated from a Nusselt number correlation somewhat like Equation 13.7, using what is called a Seider-Tate type correlation.^(13.26, 13.27)

Maximum velocity is limited by erosion, vibration and pressure drop. The maximum velocity desirable can be estimated.

$$v_{\max} = \frac{A}{\rho^{0.5}} \quad (13.16)$$

Where: v = velocity
 ρ = density
 A = factor

Metric	English
m/s	ft/s
kg/m ³	lbm/ft ³
122	100

A velocity higher than this may be used sometimes when employing an erosion resistant material with sweet fluid free of solids.

PIPE COILS

Pipe coils come in many forms. One example is the combination heat exchanger-surge tank used in small glycol dehydrators. This is simply a coil of pipe or tubing wrapped around a mandrel and inserted into a vessel. Other vessels contain pipe running longitudinally with "U" bends at the end. Banks of pipes may be placed in the water basin of a cooling tower.

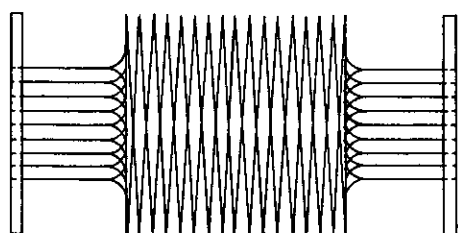
In all cases like these the liquid on the outside is flowing at a low rate, or is essentially stationary. The heat flux (heat transferred per unit time per unit area) is relatively low, but this type of exchanger is inexpensive and is suitable for some limited applications. The calculation is equivalent to that shown for other fluid exchangers.^(13.28)

SPECIAL EXCHANGERS

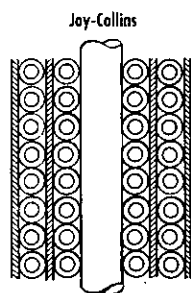
The need for close approaches and high surface areas may require exchangers possessing less weight and higher surface area per unit volume than more traditional types. At very low temperatures, a 1°C increase in approach may require an extra energy input equivalent to 55-97 kW/10⁶ std m³ of gas processed. Needed are exchangers with minimum resistance to flow, maximum surface per unit weight and volume, and low heat capacity.

Figure 13.16 shows several common types used.^(13.29) The Hampson model uses tubing wound on a mandrel. This minimizes thermal stresses, gives a large area per unit volume, and minimizes channelling on the shell side. The Trane exchanger consists of corrugated aluminum sheets brazed between flat aluminum plates in layers and sealed with aluminum channels to form the flow passages. The Joy-Collins exchanger has two (or more) concentric tubes joined together by soldering spiral-wound metal helices between their walls.

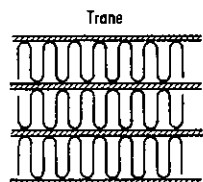
The Ramens Lamella design is a shell-and-tube type using a cylindrical bundle of flattened tubes to obtain more area per unit volume. These lamellas (flattened tubes) are welded and thus are not easy to repair



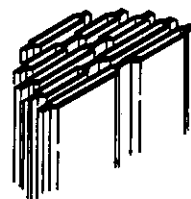
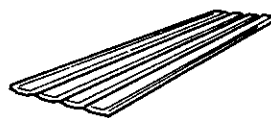
Hampson



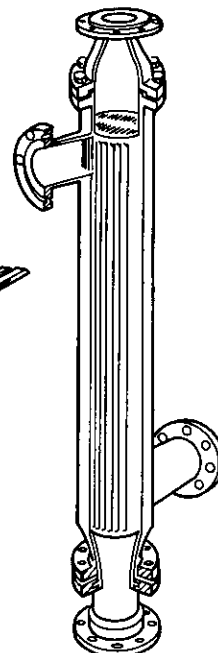
Joy-Collins



Trane

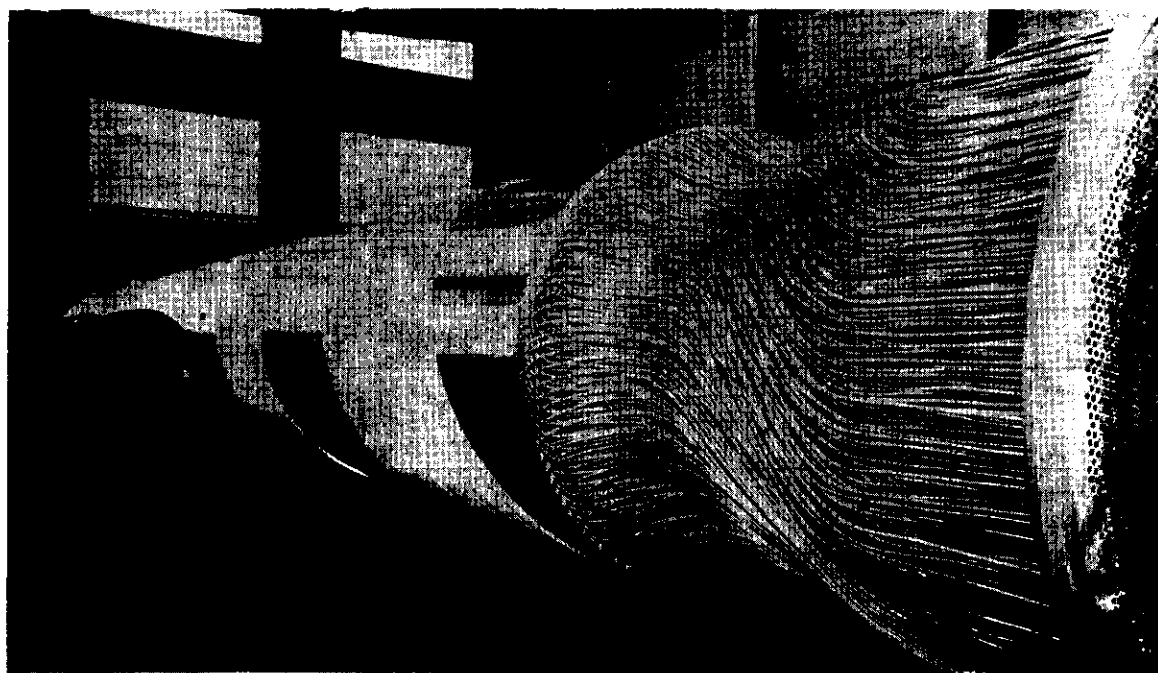


Lamella bundles



Low-Temperature Heat Exchangers

Ramens Lamella Heat Exchanger



Coil wound aluminum heat exchanger containing 10 400 m² of surface

Figure 13.16 Examples of Special Service Exchangers

or remove. Depending on metallurgy, this exchanger is available for pressure to 3.6 MPa [525 psia] and temperatures to 500-600°C [932-1112°F].

All of these may be used so that the streams involved may periodically change channels (reversing process). This is frequently used to revaporize solids frozen out of high pressure feed (such as water and CO₂) by using a lower pressure residue stream.

The coil-wound, aluminum heat exchanger also shown in Figure 13.16 has become very popular for LNG service. One main problem has been plugging from solid debris which has sometimes necessitated back-flushing with dry nitrogen or a similar material. Small tube size is the price one pays for maximum area per unit volume.

The regenerative heat exchanger of the type used in other processes also shows some promise. This may consist of one or two vessels containing a high heat capacity packing. Two vessels are required for continuous service. One is sufficient for use in intermittent processes. In operation, one stream gives up heat to the packing, which heat is, in turn, picked up by the other stream. To be effective, the two streams must have about the same enthalpy requirements and be fairly close in average temperature. Regenerative heat exchangers are fairly cheap to manufacture and may have about 6500 m² [70 000 ft²] of surface per cubic meter of volume. Several types of aluminum packing are commonly used.

RADIANT HEAT TRANSFER

This becomes a significant factor as the temperature increases. It is the major factor in most fired heaters.

The basic equation relating the radiation variables is known as the Stefan-Boltzmann Law:

$$Q = k A (T_1^4 - T_2^4) F_e F_a \quad (13.17)$$

Where:

k = constant
A = surface area
T₁ = higher temperature
T₂ = lower temperature
F_e = emissivity factor
F_a = geometry factor
Q = heat gained or lost

Metric	English
5.72×10^{-11}	1.73(E-09)
m ²	ft ²
K	°R
K	°R
—	—
—	—
kW	Btu/hr

The geometric factor accounts for relative size and shape, and distance between the two bodies exchanging radiant heat.

A given body can absorb or emit radiant energy. At thermal equilibrium the ratio of the emissive power of a surface to its absorptivity is the same for all bodies. A perfect radiator has an emissivity of one. Such a surface must have an absorptivity of one and reflectivity of zero. This hypothetical surface is commonly referred to as a *black body*. The ratio of the emissive power of an actual surface to that of a "black body" is known as the *emissivity*. At thermal equilibrium the emissivity and absorptivity of a body are identical.

Below are listed typical emissivity values for the types of surface commonly used in processing.

Surface	Emissivity
Aluminum	0.040-0.055
Iron and Steel	
Rolled sheet steel	0.66
Oxidized iron	0.74
Iron oxide (rusted surface)	0.87
Galvanized sheet iron	0.28
Brick (red)	0.93
Glass	0.94
Roofing paper (black)	0.91
Paints	
Black lacquer	0.80-0.95
Flat black	0.97
Aluminum	0.40-0.60
White enamel	0.90
Oil paints (all colors)	0.92-0.96

The above shows that aluminum surfaces will reflect more radiant heat from sun. Generally, darker, less glossy surfaces are more efficient emitters of heat. The type of surface is more important than color.

We paint primarily to protect and beautify, but in doing so it should be kept in mind that the type of surface can have a noticeable effect on heat gain or loss, particularly outdoors where rapid changes can occur between a clear sunny day and a clear black night.

Lauer has prepared a series of correlations for estimating radiant heat transfer using a pseudo-film coefficient. He equated ($hA\Delta t$) to Equation 13.17 to solve for an equivalent "h."

INDIRECT FIRED HEATERS

These are different configurations, two of which are shown in Figure 13.17. The hot combustion gas and flame heat an intermediate liquid which, in turn, heats a fluid flowing through a coil or a series of tubes. The intermediate liquid must be stable at atmospheric pressure and the maximum temperature involved. It is water, heat transfer oil or a eutectic salt, depending on temperature level. Indirect heaters have proven safe, reliable and convenient to use. Both radiation and convection are involved.

The intermediate liquid transfers heat between the fire tube and the fluid being heated by natural convection. This limits the rate of heat flux per unit area. Indirect heaters are seldom used to produce outlet fluid temperatures above 260°C [500°F]. The primary use is to heat oil and gas in production operations where the heat loads are not large. They also are used as reboilers in amine, glycol and oil stabilization units. In this case, the intermediate liquid is being boiled; there is no fluid coil or tube.

For line heaters in natural gas service typical sizing criteria are:

Fluid coil: Overall "U" (in-service) 250-300 W/m²·°C [44-53 Btu/h-ft²-°F]

Fire tube: Heat flux 28-38 kW/m² [9000-12 000 Btu/h-ft²]

For maximum firetube life it is recommended that heat flux not exceed 28 kW/m² [9000 Btu/h-ft²]

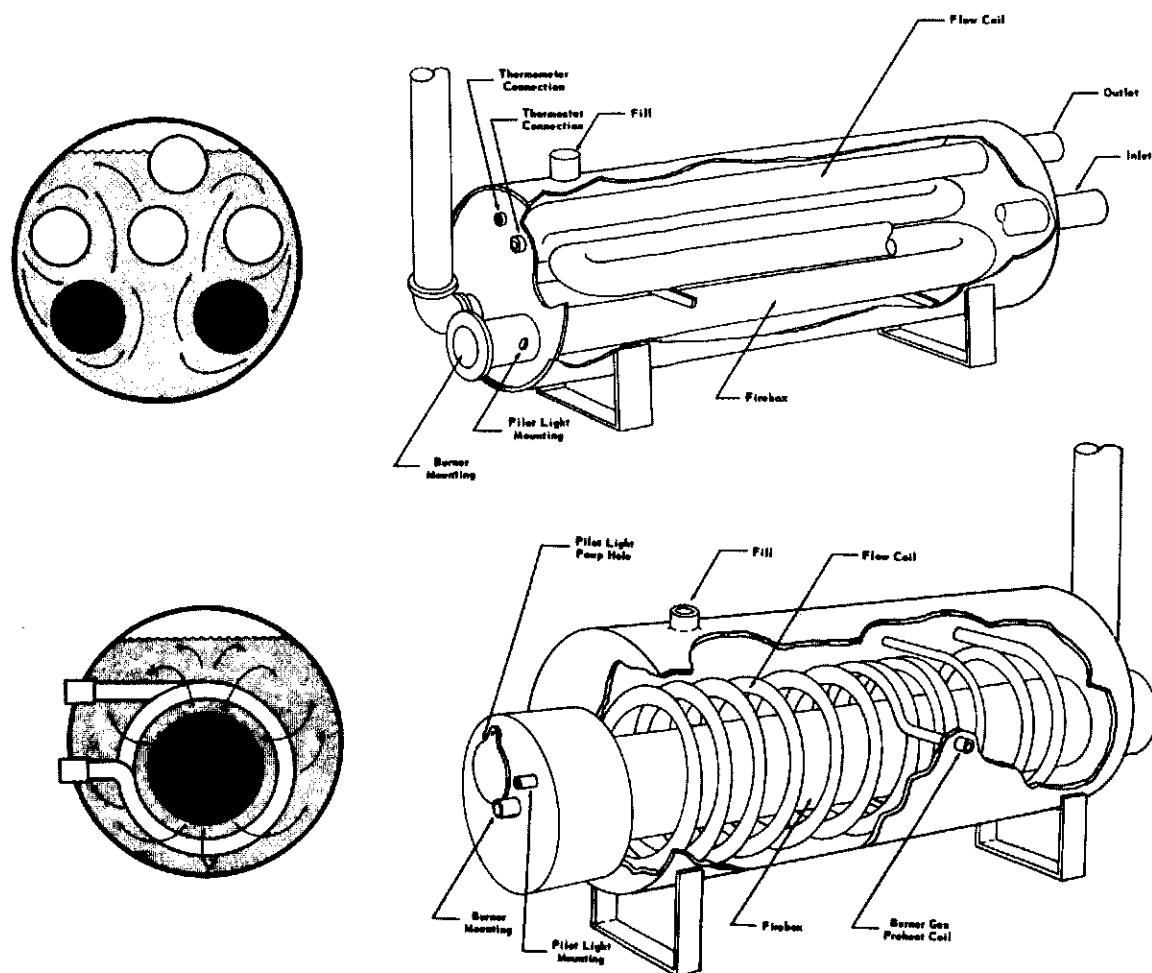


Figure 13.17 Basic Types Of Indirect Fired Heaters.

DIRECT FIRED HEATERS

Direct fired heaters are the usual equipment when large amounts of heat input are required. Modern units consist of both a radiant section and a convection section. Since the temperatures in both are very high, an intermediate heat transfer fluid may be used. A rather stable heat transfer oil, Dowtherm or some such relatively stable liquid, may be heated and then circulated to various fluid-fluid exchangers.

There is a wide variety of models and heating configurations. The choice depends on fuel cost, thermal efficiency, temperatures desired, size of the heat load and the fluid being heated. Figures 13.18 and 13.19 show various types of commonly available direct fired heaters.^(13.31)

In Figure 13.18 the fluid tubes are horizontal. The characteristics can be summarized as follows.

- A. *Cabin*. – The radiant section normally lines the walls with burners in the floor. An economical and high efficiency unit that currently is the most popular of horizontal tube units. Normal duty range: 3-30 MW [10-100 MM Btu/h].
- B. *Two-Cell Box*. – Only two boxes are shown but three or four can be used. Vertically fired from floor to give an economical, high efficiency design. Normal duty rating: 30-75 MW [100-250 MM Btu/h].
- C. *Cabin with Bridgewall*. – This divider provides two sections which can be fired individually. Can be fired horizontally or vertically. Normal duty range: 6-30 MW [20-100 MM Btu/h].

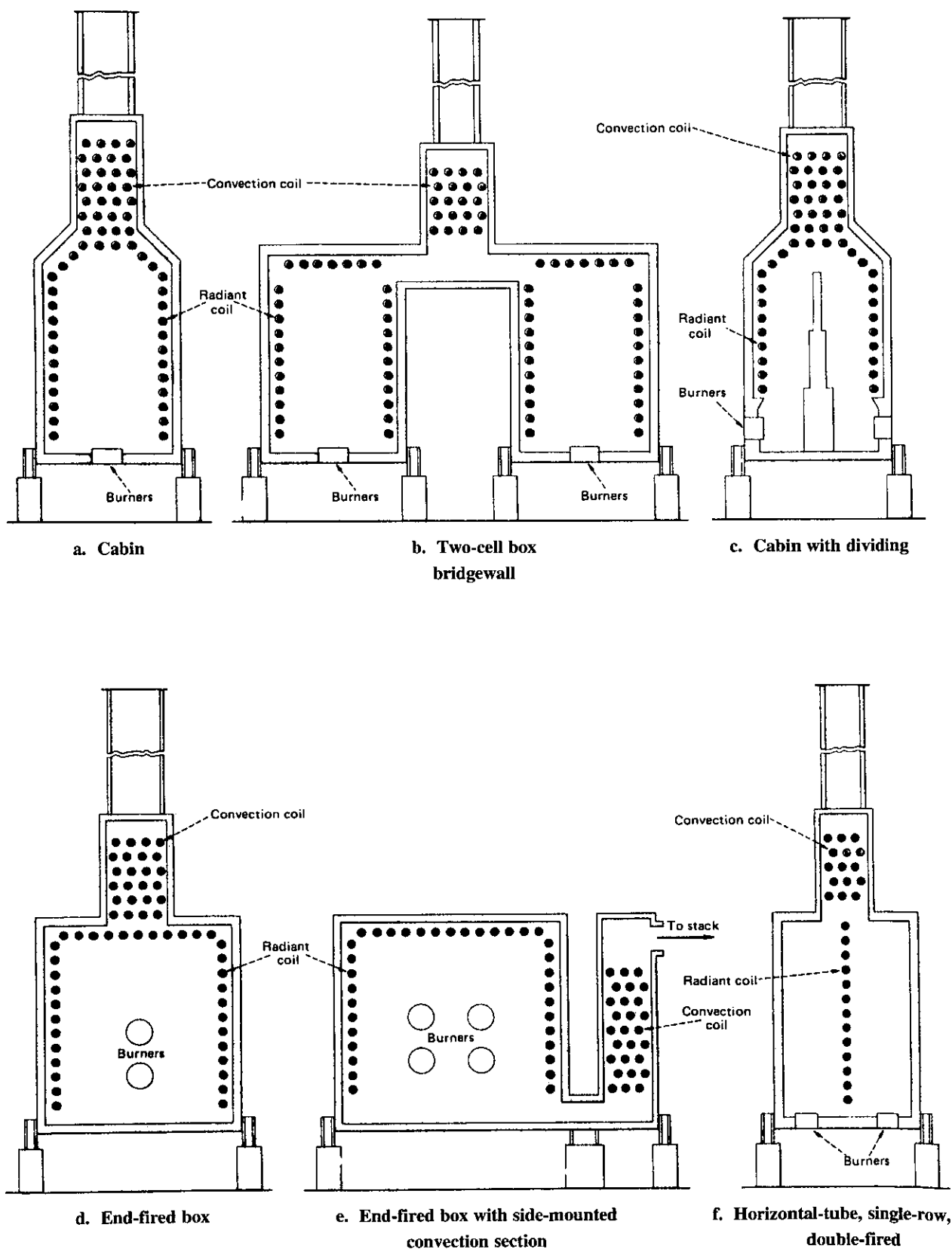


Figure 13.18 Basic Types of Direct Fired Heaters with Horizontal Tubes

- D. *End Fired Box*. – Horizontally fired as name implies. Normal duty range: 1-15 MW [5-50 MM Btu/h].
- E. *End Fired Box with Side Mounted Convection Section*. – An older type unit that may be used in new installations with high ash, poor grade fuels. More expensive design. Normal duty range: 15-60 MW [50-200 MM Btu/h].
- F. *Single Row, Double Fired*. – Consists of one or more cells shown. Often used for reactor-feed heating services. Normal duty range: 6-15 MW [20-50 MM Btu/h].

Figure 13.19 shows some comparable vertical tube heaters. The characteristics can be summarized briefly as follows.

- A. *All Radiant*. – Low cost, low efficiency design that is compact. Normal duty range 0.1-6 MW [0.5-20 MM Btu/h].
- B. *Cylindrical, Helical Coil*. – A basic low cost, low efficiency alternate to (A). Not feasible to have parallel flow coils for fluid. Normal duty range: 0.1-6 MW [0.5-20 MM Btu/h].
- C. *Cylindrical, with Cross-Flow Convection*. – This is the most popular of new vertical flow units. It is an economical, high efficiency, compact unit. Normal duty range: 3-60 MW [10-200 MM Btu/h].
- D. *Cylindrical, with Integral Convection*. – There are a large number of existing units of this type. Not often purchased now for new installation because of the limited thermal efficiency. Normal duty range: 3-30 MW [10-100 MM Btu/h].
- E. *Arbor or Wicket*. – Used most commonly for heating large quantities of gas where low pressure drop is desired. Several arbor coils may be used in one heater unit. Normal duty range: 15-30 MW [50-100 Btu/h].
- F. *Single Row, Double Fired*. – Most expensive configuration but provides high and rather uniform heat flux. Normal duty range: 6-35 MW [20-125 MM Btu/h].

Direct fired heaters have evolved over the years to reflect the need for greater efficiency and more reliable performance. With high flame temperatures and low convection film coefficients, the development of "hot spots" and tube failure always has been a problem. Tube metallurgy selection normally is a compromise between initial cost and service life. The choice of material, method of welding, configuration used, etc. must be based on experience.

Thermal efficiency is a major consideration. The major factors involved are control of excess air, additional convection heat recovery, preheating the combustion air and using turbine exhaust gas.

Excess air requires fuel to heat up ambient air to the stack temperature. Thermal efficiency is inversely proportional to excess air; stack temperature varies directly. Thus burner selection and placement (to use minimum excess air) is important.

The use of studded or fin type convection tubes has enhanced heat recovery from the hot combustion gases. The use of extended surface on the low coefficient side tends to reduce fouling inside the tubes, a possible major problem. However, said surfaces foul easier. With some fuels, control of soot is critical.

Combustion air preheat with combustion gas is now common. Thermal efficiency may reach 90% with the stack temperature being lowered to 150-175°C [300-350°F]. Preheating, though, does tend to elevate combustion temperatures which increases radiant heat transmission and tube surface temperatures.

When available, gas turbine exhaust can be a super substitute for air. It usually contains 17-18% oxygen and it already is "preheated" to 425-480°C [800-900°F]. This is a good use of waste heat.

In many cases, turbine or engine combustion gas can be flowed through a convection section to heat process streams. This heat can be used directly or indirectly to generate steam, run reboilers, regenerate solid desiccant units, heat gas and liquid streams, etc.

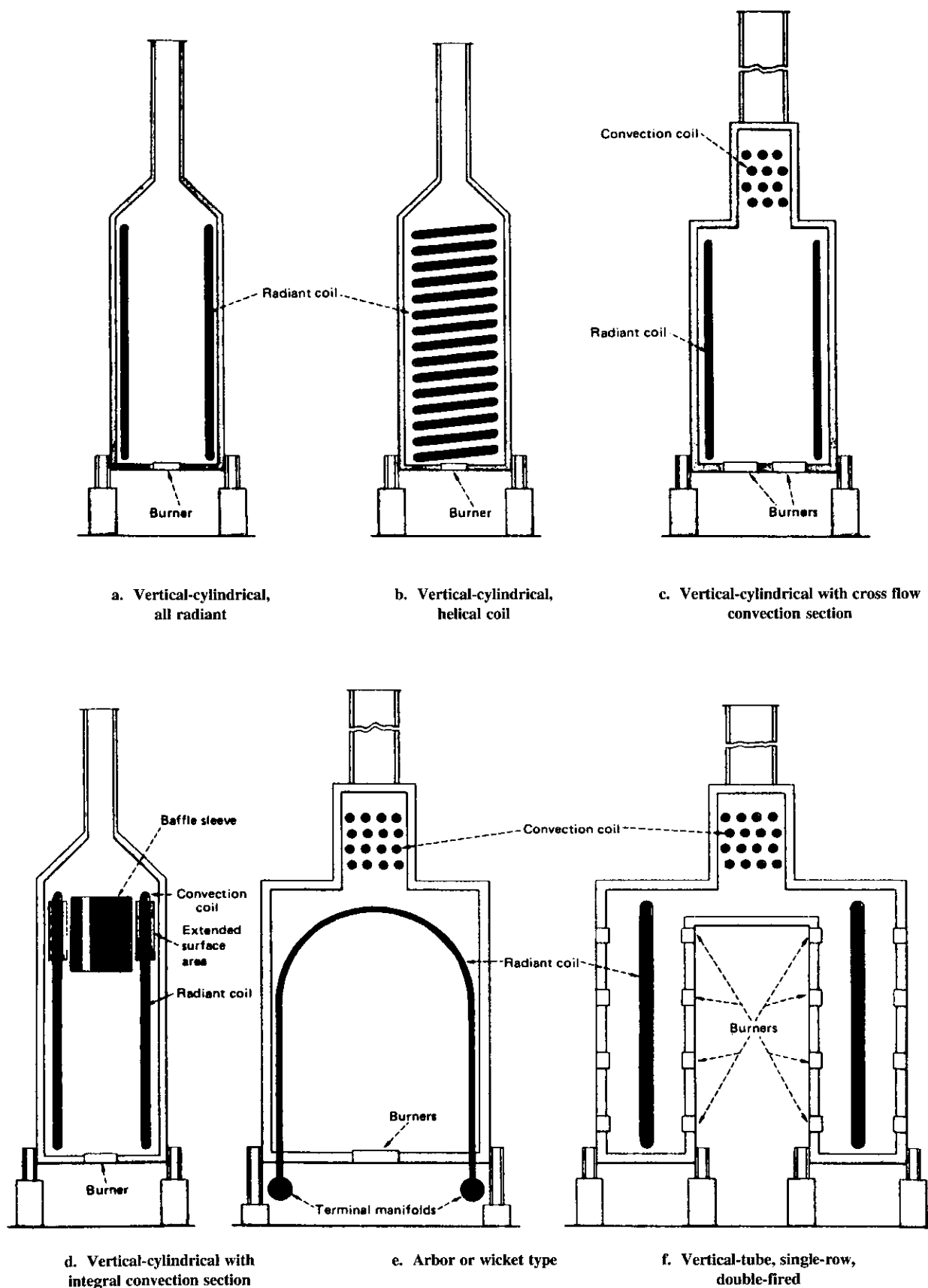


Figure 13.19 Basic Types of Direct Fired Heaters with Vertical Tubes

Sufficient purge must precede the lighting of the pilot (if it is not operated continuously). This is followed by automatic pilot ignition and use of a flame scanner to confirm ignition. Flow in the tubes must commence before the main burner(s) light. Said burners also must shut off if flow reaches some minimum level or stops completely (on purpose or accidentally). Part (a) of Figure 13.20 shows the basic design on the dehydration system.

Part (b) shows the control scheme for the four concerns shown. Control of maximum firing rate limits the temperature fluctuations. The firing valve, control signal line is connected to a low-select relay with a dual input from the temperature controller and from a regulated preset source that controls heat level.

Over-temperature is prevented by a pressure switch on the temperature transmitter output to the controller. This switch shuts off the main burner to limit maximum gas temperature.

When the burner turns off, the tubes and cabin are still hot. The *heat removal cycle* flows fluid for 710 minutes after shut-off to cool down the unit. This minimizes rapid thermal stresses.

Shut-down controls are critical. They must shut off process flow entering, back flow of the process fluid leaving and, of course, shut off burner fuel. For tube failure a pressure switch in the heater cabin may be more practical than an excess flow valve on the process fluid. One also may wish to have a high temperature shut-down in the stack.

In some cases, external infrared scanners are used to shut down a heater. All shut-down valves should be far enough from the heater to allow safe manual closure in the event of control failure.

COOLING WITH AIR

Air is a convenient cooling medium; there is so much of it and it is everywhere. The air can be used directly in an *aerial cooler*, used to cool water in a *cooling tower* or in a combined air-water cooler. The latter is a special unit in which the air entering an aerial cooler has been precooled by a water spray. It usually is used where water is in short supply and ambient air temperatures are high.

Table 13.2 summarizes the pros and cons of air and water cooling.(13.41) As a general rule, air cooling is preferred if the ambient air temperature is low enough to provide efficient cooling. Cooling towers have almost no application in offshore, arctic or humid locations.

COOLING TOWERS

The sketch below summarizes the process involved. It is possible to cool the water in the tower only when the entering air is unsaturated (above its dewpoint temperature). In the process of passing unsaturated air countercurrent to the warm water entering, some of the water evaporates. The latent heat of that water evaporating must be supplied from somewhere. Most comes from cooling that water, not evaporating; some is supplied by the air. This is the mechanism by which water is cooled – evaporating a portion to cool the remainder. The maximum amount evaporated is limited by the air's capacity for water. The actual amount evaporated will depend on the effectiveness of the mass transfer (efficiency of air-water contact, area of contact, distribution, etc.). The driving force is water concentration difference. The resisting force is a function of effective area of air-water contact.

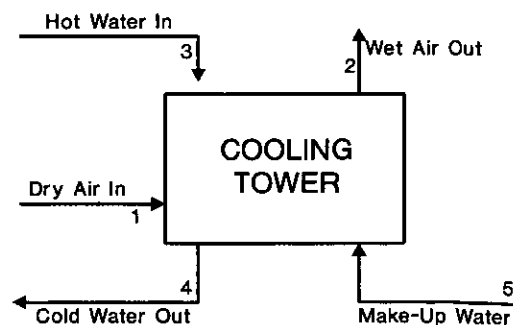


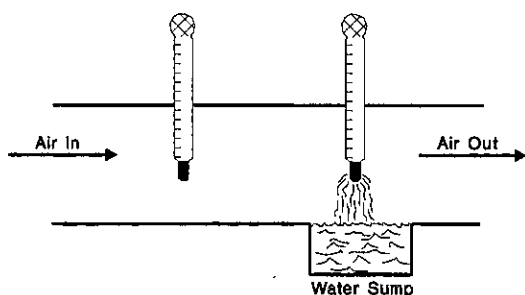
TABLE 13.2
Comparison of Air and Water Cooling

In Favor of Water Cooling	
Air Cooling	Water Cooling
Because of air's low specific heat, and a dependence on the dry-bulb temperature, air cannot usually cool a process fluid to low temperatures.	Water can usually cool a process fluid from 10°F to 5°F lower than air, and recycled water can be cooled to near the wet-bulb temperature of the site in a cooling tower.
Air coolers require large surfaces because of their low air-film heat-transfer coefficients and the low specific heat of air.	Water coolers require much less heat-transfer surface in compact, well-proven exchanger designs.
The seasonal variation in air temperatures can affect performance, while rain and sun can cause appreciable variations in the daily temperature. If winter temperatures are low, they may cause process fluids to freeze.	Water is less susceptible to temperature variations.
Air coolers should not be located near large obstructions such as buildings, trees, etc., since air recirculation can set in.	Water coolers can be located among the other equipment.
Air coolers require finned tubes—a specialized technology.	Well-established designs of shell-and-tube exchangers satisfactory.
In Favor of Air Cooling	
Air Cooling	Water Cooling
Air is available free, with no preparation costs.	Water for cooling is generally scarce; and when it is available, it must be brought to the site by pump, pipeline, or from a well, etc., at an attendant cost.
Plant location is not restricted by air cooling.	Sites for large plants in particular are dependent on suitable sources for water cooling.
Air is seldom corrosive, so that less provision need be made for fouling and cleaning.	Water is corrosive and requires treatment to control both scaling and deposition of dirt.
Operating costs for air coolers are lower, since the draft losses are on the order of 0.5-1.0 in. of water.	Operating costs for water coolers are higher, because the cooling-water circulation pump can have a head running to tens of feet of water, depending on the location of the cooler and the cooling towers.
There is less danger of contaminating the cooling medium with air cooling.	In many processes where a toxic fluid needs cooling, there is danger of contaminating the cooling water.
Maintenance costs for air-cooling systems are generally 20 to 30% those of water cooling systems.	Maintenance of cooling water is costly, because: there is more equipment; water contains living organisms that grow in the warm conditions and thus foul exchangers; minerals such as iron can deposit on tubes as oxides or hydroxides; the inside of the tubes (water side) requires shutdown and dismantling of the exchanger for cleaning.

The following quantities must be known or specified for cooling tower design:

Dry Bulb. – The maximum design temperature of the air as measured by a temperature recording device.

Wet Bulb. – The design temperature established by passing air over a thermometer whose bulb is covered with wicking saturated with water. This may be established using a *sling psychrometer* or a device of the type shown at left. Some of the water on the wick evaporates as the unsaturated air passes over it, cooling the bulb. However, following cooling, heat is transferred to the bulb by the thermometer stem and the air. If the process is continued for several minutes, at high enough air rates, a temperature equilibrium is established. This is the wet bulb, the minimum temperature to which the water could be cooled in a cooling tower.



Dewpoint. – The temperature at which the air is saturated with water at the pressure involved (atmospheric pressure).

Humidity Ratio (ω). – Ratio of the mass of water vapor (M_v) to the mass of dry air (M_a) in an air-water vapor mixture.

Relative Humidity. – The ratio of the partial pressure of the water vapor in the air mixture (P_p) to the saturation (vapor) pressure (P_v) at the same temperature. The relative humidity (ϕ) is equal to 1.0 at saturation (dewpoint) conditions. " ϕ " is related to " ω " by the equation

$$\phi = \frac{P_p}{P_v} = \frac{\omega P_a}{0.622 P_v} \quad (13.18)$$

Where: $\omega = (0.622 P_p)/P_a$ (for ideal gases)
 P_a = partial pressure of the dry air at total pressure P

Range. – Established by the tower heat load and the water circulation rate. It is a measure of tower loading. The most common range is 5-20°C [9-36°F]. Both range and water circulation rate have a significant effect on tower size.

The equation for calculating range is

$$\text{Range} = \frac{\text{Heat Load}}{(\text{Water Rate})(\text{Water } C_p)} \quad (13.19)$$

Approach. – The temperature difference between the cold water leaving the tower and the wet bulb temperature. Approach usually has the greatest effect on tower size and cost. An 8-9°C [14-16°F] approach is very standard.

Tower cost depends on the approach and the wet bulb temperature. The table below is based on an approach of 9°C [16°F] which means that the outlet water temperature shown is 9°C [16°F] above the wet bulb.

Outlet Water Temperature	Relative Cost
32°C	1.0
30	1.1
25	1.35
20	2.0

Thus, as the wet bulb decreases, to keep the approach the same results in higher cost. The wet bulb seldom exceeds 30°C [86°F]. For most areas of the world it is often 24°C [16°F] or less.

It is customary to specify a dry bulb and wet bulb that will not be exceeded over 5 percent of the time. Designing for a higher temperature is very expensive compared to the benefit.

Energy Balance

In a tower operating at steady state conditions, enthalpy input = enthalpy output. The following overall energy balance may then be written.

$$Q = (M_L)(4.19)(t_3 - t_4) = M_a [(0.92)(t_2 - t_1) + (\omega_2 h_{w2} - \omega_1 h_{w1})] \quad (13.20)$$

Where: ω = kg water vapor per kg dry air
 h = enthalpy, kJ/kg
 M = kg per hour

Subscripts: 1 = inlet conditions for air
 2 = outlet conditions for air
 a = dry air
 L = liquid water

3 = hot water in
 4 = cool water out
 w = water vapor

Equation 13.20 assumes the air is at, or near, atmospheric pressure. For a fixed set of temperature conditions, this equation may be solved for " M_a ," the necessary flow rate to produce these temperatures. This equation is possible because there is no change in the mass of dry air in or out, or process water in or out. The make-up water preserves the latter equality. The make-up water requirements are found by a water balance, which reduces to

$$M \text{ (kg/h)} = M_a (\omega_2 - \omega_1) \quad (13.21)$$

Equation 13.20 simply says that in an adiabatic tower, the energy lost by the water equals the energy gained by the air stream. The water makeup (including driftage loss) is typically 2-5% of the circulation rate.

PSYCHROMETRIC CHART

In solving some of the previous equations a psychrometric chart like Figure 13.21 is useful. The saturation curve is the dewpoint temperature at 100 kPa pressure of saturated air.

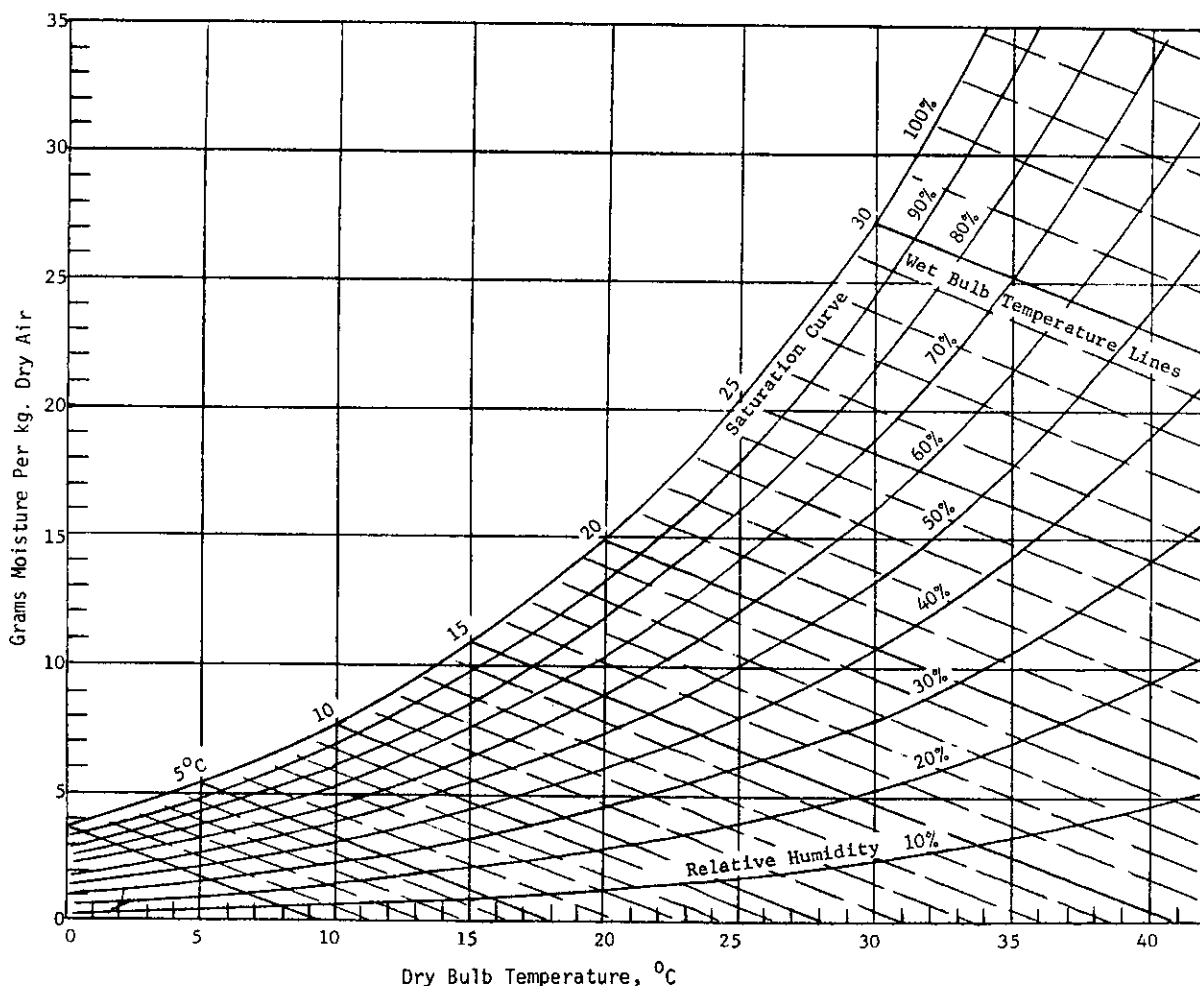


Figure 13.21 Psychrometric Chart Barometric Pressure: 100 kPa

The wet bulb and dry bulb obtained from a psychrometer test fixes a point on this chart. From this point one can find the moisture content of the air and the relative humidity.

Example 13.4: From a test, the Dry Bulb temperature = 25°C and the Wet Bulb temperature = 21°C. What is the relative humidity and moisture content of the air?

From Figure 13.21, Rel. Hum. (ϕ) = 70%, Moisture content = 14g/kg air

The dewpoint of unsaturated air is found by reading horizontally from the point fixed by the wet bulb and dry bulb to the dewpoint (saturation) curve. For the above example, the dewpoint equals 19°C.

If we use a 9°C [16°F] approach to a WB of 21°C [70°F], the cold water should leave the tower at 30°C [86°F]. If this cold water is used in a heat exchanger where the approach is 10°C [19°F] to cool oil, the oil will leave at 40°C [104°F]. So ... knowing the wet bulb and the two approaches involved, one can predict what may be accomplished from the cooling tower.

The values in Figure 13.21 are dependent on pressure and only apply at 100 kPa [14.5 psia]. One can correct for pressure by using ideal gas laws. In many cases this correction is negligible.

Practical Tower Choice

There are five basic types of cooling towers – atmospheric spray, counter-flow, cross-flow, hyperbolic natural draft, and coil-shed. Air is moved by natural air movement, induced draft (fans suck air through), or forced draft (fans blow air through).

The atmospheric spray tower is normally used only for small installations. It is seldom recommended for gas process purposes. A counter-flow tower has a small base area and might be preferred for oil-contaminated or heavily sedimented waters. Except for these cases, the cross-flow tower is usually preferred. It gives low draft loss, may use a higher water loading, a longer "range," and achieve closer approaches with minimum fan horsepower requirements. A cross-flow tower also requires a lower pumping head and does not require nozzles (using a fill material instead).

Hyperbolic towers are used primarily in the electric power industry in generator stations. They offer favorable economics in very large installations.

A coil-shed tower is simply a cross-flow tower with coils in the base. Rather than circulating water from a basin to external coolers, said cooling is done in the tower proper. They are very flexible and may be used in combination with external coolers for multiple cooling functions.

Induced draft is usually preferred over forced draft because of a tendency for the latter to recirculate wet exit gas back into the inlet air stream. This adversely affects tower performance by increasing the inlet air wet bulb. The fan horsepower requirements may be estimated for both types.

There are many mechanical considerations too numerous to discuss in detail here. One is the choice of materials. Redwood has been the traditional material but is rapidly being replaced by metals and plastics. Fiber glass reinforced plastic has gained wide acceptance. It is lighter than redwood and the cost is generally competitive. Fire retardant plastics cost about 50-60% more. Even stainless steel has been used successfully. The initial cost is higher but it can show real advantage if long service life, with minimum maintenance cost, enters the picture.

Data are readily available from manufacturers and in the literature for the general sizing and specification of cooling towers. Short-cut analyses provide convenient input for planning purposes.^(13.34,13.35)

Water Treatment

Environmental regulations have altered water treatment practices. No longer may one dump water containing treatment chemicals.

Water treatment involves contending with the following impurities:

1. Salts like calcium and magnesium that form scales
2. Silica which also forms a scale
3. Iron which can result in corrosion promoting deposits
4. Dissolved gases that can be corrosive
5. Oil which reduces heat transfer, interferes with corrosion inhibitors and accelerates the growth of microorganisms
6. Microbiological species that cause a host of problems

Details of water chemistry that affect cooling tower water treatment are shown in Reference 13.36.

Addressing these problems requires (1) control of pH, (2) attention to the level of total dissolved solids (TDS), and (3) using an effective corrosion inhibition program. Figure 13.22 shows a water treatment control system.^(13.37) Part (a) shows the sensing and injection system in detail. Part (b) shows the control system located on a cooling loop around a water condenser being serviced by cooling tower water. Chemical injection in a loop promotes good mixing and accurate control of concentration.

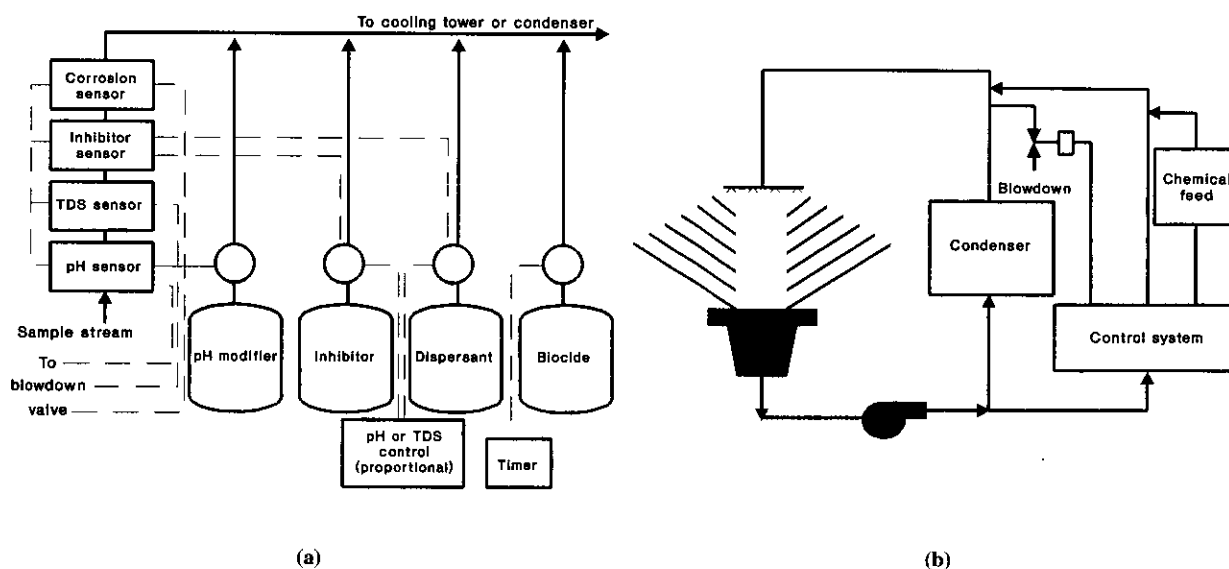


Figure 13.22 Control of Water Quality in a Cooling Tower

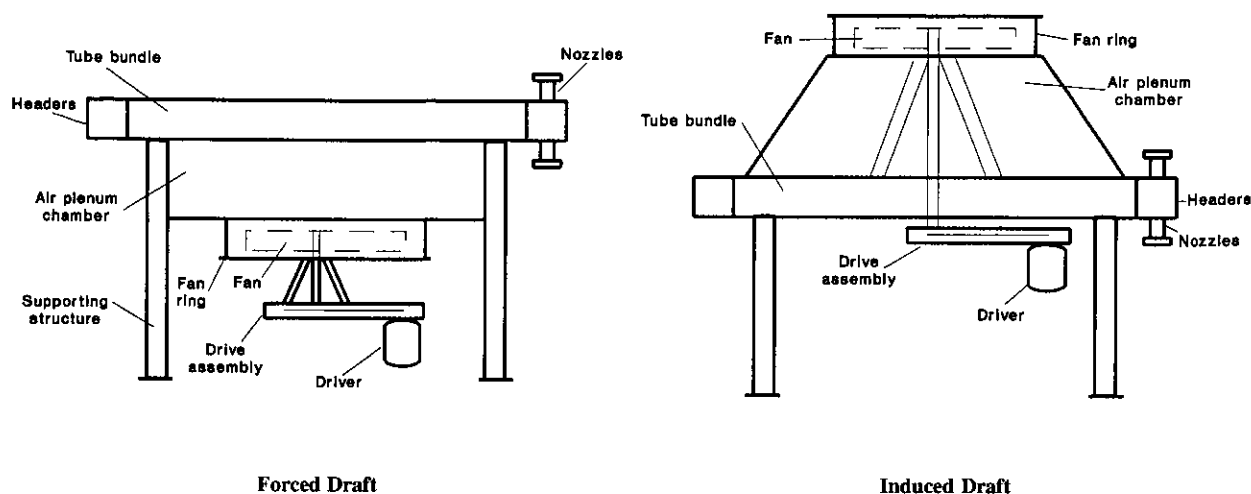
The use of automatic monitors and control is convenient but one must recognize that these need regular maintenance and calibration to be reliable. They minimize but do not eliminate the need for routine human attention. The operating cost for labor and chemicals may be very large.

AERIAL COOLERS

The aerial cooler is very popular even where water for cooling is available. It is mechanically simple, flexible and eliminates the nuisance and cost of water treating. However, it also has several disadvantages. In warm climates it is not capable of producing as low a temperature as water. This can increase the capital

cost of the system substantially by lowering thermal efficiency. If water is available a very careful study should be made of the cooling alternatives. This study should not merely compare the cost of the cooling system but should examine the cost of the alternative on the total system.

Figure 13.23 is a schematic view of the two basic types of aerial coolers. The forced draft type is cheaper. However, many prefer the induced draft type on the basis it is more efficient.



Courtesy the Rainey Corp.

Figure 13.23 The Two Basic Types of Aerial Coolers

With the forced draft type there is a possibility that hot air leaving the top will flow around the unit and be drawn through again. Any by-passed air will be hotter than the ambient air and will lower the cooling efficiency. This by-passing of hot air is less likely with the induced draft type because air is exhausted at higher velocity through the reduced area of the fan ring and is more likely to be dissipated by normal winds.

Air is abundant but has a heat transfer effectiveness of about 1-2% compared to water. The use of fins on the air side makes air about 20% as effective as water, if the fouling effect of water on exchange is ignored. With the type of water often used, air may be 50-70% as effective as water.

Historically, many vendors have compromised on design to minimize initial cost. At this point in time, energy consumption is equally important. One study of aerial coolers (purchased as quoted) showed that two-thirds performed at less than design capacity. About 75% of these had insufficient air flow. About one-half of the units were within 10% of quoted performance. These results may not be typical of all units, but these units were purchased by a well engineered, major oil company. I mention this to point out that a "guarantee" is no substitute for careful customer appraisal.

The following data sheet illustrates the variables that are involved in designing and specifying an aerial cooler. One must consider the cooling tube design, fan arrangement, number of transfer bays, control of cooled fluid temperature as the ambient air temperature changes and the usual mechanical considerations.^(13.38-13.42)

Design of an optimum aerial cooler is a trial-and-error calculation that should not be undertaken by a novice. There are many combinations of air flow rate, tube design, type of fins, and fan type and speed. For a given cooling service, the designer must assume an air flow rate. Increased rate increases U, LMTD, pressure drop and fan power requirements. For each rate there is a further change with tube diameter, pitch and fin type. So, multiple calculations are required to truly optimize design.

DATA SHEET - AIR COOLED HEAT EXCHANGER Sheet of

1 CUSTOMER		EQUIPMENT TAG NO.	
2 PLANT LOCATION		MANUFACTURER	
3 SERVICE		TYPE: INDUCED/FORCED DRAFT	
4 SIZE		NO. OF BAYS	
5 SURFACE PER UNIT - FINNED TUBE		FT ² BARE TUBE	
6 HEAT EXCHANGER		STUHM (STD (EFF))	
7 TRANSFER RATE - FINNED TUBE		BARE TUBE SERVICE	
		CLEAN	
		STUHM, FT ² /hr	
PERFORMANCE DATA			
TUBE SIDE			
10 FLUID CIRCULATED		GRAVITY, LBS	
11 TOTAL FLUID ENTERING		FOUR POINT	
12		BUBBLE POINT	
13 TEMPERATURE		DEN POINT	
14 LIQUID		SPECIFIC HEAT	
15 VAPOR		LATENT HEAT	
16 INCORPOR		COND. THERMAL	
17 STAY		OUTLET PRESSURE	
18 WATER		ALLOWABLE PRESSURE DROP	
19 VISCOSITY (LIQ/VAP)		CALC. PRESSURE DROP	
		FOULING RESISTANCE, INSIDE	
AIR SIDE			
22 AIR QUANTITY		ALTITUDE	
23 AIR QUANTITY/FAN		TEMPERATURE IN	
24 STATIC PRESSURE		TEMPERATURE OUT	
25 FACE VEL		MINIMUM AMBIENT	
DESIGN - MATERIALS - CONSTRUCTION			
27 DESIGN PRESSURE		DESIGN TEMP.	
28 TUBE BUNDLE		TUBE	
29 NO. TUBE ROWS		MATERIAL	
30 NO. BAY		ASTM	
31 ARRANGEMENT		OD. IN. MIN. THICK.	
32 BUNDLE IN PARALLEL		NO. BUNDLE	
33 BAY IN PARALLEL		PITCH	
34 BUNDLE FRAME		GASKET MATERIAL	
35 MISCELLANEOUS		CORROSION ALLOW.	
36 STRUCTURE WITH CORROSION		PIPE JOINT MODEL	
37 SURFACE PREPARATION		PIPE JOINT MODEL	
38 LOUVER		RATING & FACTOR	
39 CODE - ASME VIII		STAMP - VES/NO	
MECHANICAL EQUIPMENT			
41 FAN		DRIVER	
42 MFD. & MODEL		TYPE	
43 NO. BAY		NO. BAY	
44 DIAMETER		RPM	
45 NO. BLADES		PITCH	
46 BLADE MATERIAL		ANGLE	
47 MIB MATERIAL		MFR.	
48 CONTROL ACTION ON AIR FAILURE		FAN PITCH	
49 NOTES - GIVE TUBE COUNT OF EACH PASS WHEN IRREGULAR		LOUVERS	
50		OPEN/CLOSED	
51			
52 PLOT AREA		PROPOSAL ORG. NO.	
53 Prepared:		Approved:	
Date:		Rev.	

ADVANTAGES OF INDUCE DRAFT DESIGN

1. Easier to shop assemble, ship and install.
2. The hoods offer protection from weather.
3. Easier to clean underside when covered with lint, bugs, debris.
4. More efficient air distribution over the bundle.
5. Less likely to be affected by hot air recirculation.

DISADVANTAGES OF INDUCED DRAFT DESIGN

1. More difficult to remove bundles for maintenance.
2. High temperature service limited due to effect of hot air on the fans.
3. More difficult to work on fan assembly, i.e. adjust blades due to heat from bundle, and their location.

ADVANTAGES OF FORCED DRAFT DESIGN

1. Easy to remove and replace bundles.
2. Easier to mount motors or other drivers with short shafts.
3. Lubrication, maintenance, etc. more accessible.
4. With reinforced straight side panels to form a rectangular box type plenum, shipping and mounting is greatly simplified, permitting complete preassembled shop-tested units.
5. Best adapted for cold climate operation with warm air recirculation.

Disadvantages of the force draft design are the list of "advantages of the induced draft design."

Heat Transfer Calculations

The basic calculation approach is the same as other exchangers. Table 13.3 shows a group of overall heat transfer coefficients based on bare tube area.^(13.38) These are useful as a first step in planning before choosing a particular fin type on the outside of the tube.

The optimum air temperature rise across the tubes may be estimated by the equation

$$(t_2 - t_1) = (0.005)(U) \left[\frac{(T_2 + T_1)}{2} - t_1 \right] \quad (13.22)$$

Where:

- t_2 = outlet air temperature
- t_1 = inlet air temperature
- T_2 = temperature of process fluid out
- T_1 = temperature of process fluid in
- U = value from Table 13.3

Metric	English
°C	°F
°C	°F
°C	°F
°C	°F
Btu/hr-ft ² -°F	

The optimum air temperature rise is also a function of the range ($T_1 - T_2$) of the process fluid. The value of $(t_2 - t_1)$ calculated from Equation 13.22 should be corrected using the equation

$$CF = 0.89 + A (T_1 - T_2) \quad (13.23)$$

Where:

- CF = correction factor
- A = constant

Metric	English
dimensionless	
0.0025	0.0014

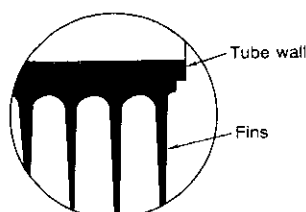
For a specified cooling load and conditions, the outlet air temperature can be estimated. From this an LMTD can be found to calculate bare tube area. Figures 13.11 and 13.12 give "F," the LMTD correction factor.

TABLE 13.3
Typical Overall Heat Transfer Coefficients

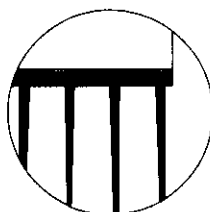
Liquid Coolers				Condensers	
Material	Heat-transfer coefficient, Btu/h-ft ² -°F	Material	Heat-transfer coefficient, Btu/h-ft ² -°F	Material	Heat-transfer coefficient, Btu/h-ft ² -°F
Oils, 20° API:		Heavy oils, 8-14° API:		Steam	140-150
200°F avg. temp.	10-16	300°F avg. temp.	6-10	Steam -	
300°F avg. temp.	13-22	400°F avg. temp.	10-16	10% noncondensibles	100-110
400°F avg. temp.	30-40	Diesel oil	45-55	20% noncondensibles	95-100
Oils, 30° API:		Kerosene	55-60	40% noncondensibles	70-75
150°F avg. temp.	12-23	Heavy naphtha	60-65	Pure light hydrocarbons	80-85
200°F avg. temp.	25-35	Light naphtha	65-70	Mixed light hydrocarbons	65-75
300°F avg. temp.	45-55	Gasoline	70-75	Gasoline	60-75
400°F avg. temp.	50-60	Light hydrocarbons	75-80	Gasoline-steam mixtures	70-75
Oils, 40° API:		Alcohols & most organic solvents	70-75	Medium hydrocarbons	45-50
150°F avg. temp.	25-35	Ammonia	100-120	Medium hydrocarbons	
200°F avg. temp.	50-60	Brine, 75% water	90-110	w/ steam	55-60
300°F avg. temp.	55-65	Water	120-140	Pure organic solvents	75-80
400°F avg. temp.	60-70	50% ethylene glycol & water	100-120	Ammonia	100-110

Vapor Coolers					
Material	Heat-transfer coefficient, Btu/h-ft ² -°F				
	10 psig	50 psig	100 psig	300 psig	500 psig
Light hydrocarbons	15-20	30-35	45-50	65-70	70-75
Medium hydrocarbons & organic solvents	15-20	35-40	45-50	65-70	70-75
Light inorganic vapors	10-15	15-20	30-35	45-50	50-55
Air	8-10	15-20	25-30	40-45	45-50
Ammonia	10-15	15-20	30-35	45-50	50-55
Steam	10-15	15-20	25-30	45-50	55-60
Hydrogen - 100%	20-30	45-50	65-70	85-95	95-100
- 75% vol.	17-28	40-45	60-65	80-85	85-90
- 50% vol.	15-25	35-40	55-60	75-80	85-90
- 25% vol.	12-23	30-35	45-50	65-70	80-85

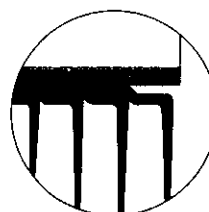
Tube sizes can vary from 15.9-38.1 mm [5/8-1 1/2 in.], but the standard size is 25.4 mm [1 in.]. Tube layout is triangular. Tube pitch is the minimum which avoids fin contact or overlap.



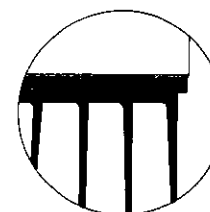
Extruded



Embedded



Double Footed



Footed

The types of fins vary with the service. They are either tension wrapped, solder bonded or extruded. The latter are the most expensive. Fin height varies from 1/2-5/8 in. and normally there are 8-11 fins per inch.

The air-side film coefficient for a typical fin tube (based on extended area) can be estimated from the equation

$$h_a = \frac{A (v_g)^{0.6}}{d^{0.3}} \quad (13.24)$$

Where: h_a = air-side film coefficient
 v_g = air velocity by tubes
 d = outside diameter of bare tubes
 A = constant

Metric	English
$\text{kJ/h} \cdot \text{m}^2 \cdot ^\circ\text{C}$	$\text{Btu/hr} \cdot \text{ft}^2 \cdot ^\circ\text{F}$
m/s	ft/sec
cm	in.
92.1	1.67

This "h" would be used with other data in Equations 13.3 or 13.4 to find an overall "U" for comparison with values in Table 13.3.

Fans

The fan power requirements can be estimated from the equation

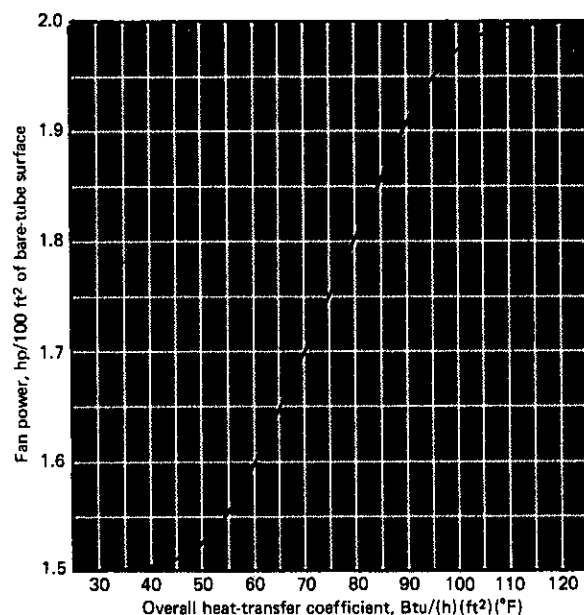
$$\text{kW} = \frac{(\Delta P_a)(Q_a)}{(A)(\text{Efficiency})} \quad (13.25)$$

Where: Q_a = air flow rate
 ΔP_a = air pressure drop in cooler
 A = constant

Metric	English
m^3/s	ft^3/sec
kPa	in. H_2O
1.0	142

Efficiency varies from 0.4-0.75; 0.7 is a useful planning number

Pressure drop varies with air rate, tube diameter, pitch and number of tube rows. For planning purposes a pressure drop of 25 Pa [0.10 in H_2O] per tube row can be used. For most gas processing applications the number of tube rows varies from 3-6.



The following figure is a rough estimate of horsepower based on the value of "U" from Table 13.3 for bare tubes. The number of fans and heat transfer bays will vary with the installation. Table 13.4 can be used to estimate exchanger size. Two fans/unit are generally preferred because of the additional flexibility in controlling air flow.

Noise control is a serious concern. A common specification is that fan and motor noise shall not exceed 85-90 dBA at a distance of three feet from the fan ring. One can estimate the sound pressure level by the equation

$$\text{dBA} = 65 + 30 (\log V) + 10 \log (\text{hp}) + 20 (\log d) \quad (13.26)$$

Where: dBA = relative sound level in decibel
log = logarithm to base 10
 V = fan tip speed. (0.001)(ft/min.)
hp = fan horsepower
 d = fan diameter, ft.

TABLE 13.4
Approximate Bare Tube Area Versus Unit Size^(13.38)

1-in. O.D. bare tube on 2 ^{3/8} -in. Δpitch						
Approximate unit width, ft.	Tube length, ft.	Fans per unit	No. of tube rows in depth			
			3	4	5	6
4	4	1	49	64	81	97
	6	1	73	97	122	146
	8	2	98	129	163	194
	10	2	123	162	204	243
6	6	1	121	160	201	240
	8	1	161	213	268	320
	12	2	242	320	402	481
	14	2	282	374	469	561
8	8	1	224	297	373	446
	10	1	280	372	466	558
	12	1	336	446	559	669
	14	1	392	520	652	781
	16	2	448	595	746	892
	20	2	560	744	932	1116
10	24	2	672	892	1119	1339
	10	1	351	466	584	699
	12	1	421	559	701	839
	14	1	491	652	817	979
	16	1	561	746	934	1119
	20	2	702	932	1168	1399
12	24	2	842	1119	1402	1678
	30	2	1053	1399	1752	2098
	32	2	1123	1492	1869	2238
	12	1	515	685	858	1028
	14	1	601	799	1001	1199
	16	1	687	913	1144	1370
14	20	1	859	1142	1430	1713
	24	2	1031	1370	1716	2056
	30	2	1289	1713	2145	2570
	32	2	1374	1827	2288	2741
	36	2	1546	2056	2574	3084
	40	2	1718	2284	2861	3426
16	14	1	700	931	1166	1397
	16	1	800	1064	1333	1597
	20	1	1000	1330	1666	1996
	24	2	1201	1597	1999	2395
	30	2	1501	1996	2499	2994
	32	2	1601	2129	2666	3194
18	36	2	1801	2395	2999	3593
	40	2	2001	2661	3332	3992
	16	1	897	1190	1492	1785
	20	1	1121	1488	1865	2232
	24	1	1345	1785	2238	2678
	30	2	1682	2232	2798	3348
20	32	2	1794	2381	2984	3571
	36	2	2018	2678	3357	4018
	40	2	2242	2976	3730	4464
	20	1	1247	1655	2075	2483
	24	1	1496	1987	2490	2980
	30	2	1870	2483	3112	3725
22	32	2	1995	2649	3320	3974
	36	2	2244	2980	3735	4470
	40	2	2494	3311	4150	4967
	20	1	1404	1865	2337	2798
	24	1	1685	2238	2804	3357
	30	2	2106	2798	3505	4197
24	32	2	2246	2984	3739	4477
	36	2	2527	3357	4206	5036
	40	2	2808	3730	4674	5596

Notes:

1. Assume 4 rows of tubes in depth except for the following condition:
 - a. If the temperature range on the process side is 10°F or less, assume 3 rows.
 - b. If the temperature range of the process fluid falls between 10°F and 20°F, and special materials of construction are required, assume 3 rows.
 - c. If the temperature range of the process fluid is between 100°F and 200°F and/or the assumed overall heat-transfer rate is less than 60, assume 5 rows.
 - d. If the temperature range of the process fluid is between 200°F and 300°F and/or the overall heat-transfer rate is less than 40, assume 6 rows.
 - e. If the temperature range of the process fluid is greater than 300°F and/or overall heat-transfer rate is less than 30, assume 8 rows.

2. Relative to 14 BWG, the effect of tube-wall thickness on cost is:

Average gage	Cost factor
12 Bwg	1.025
14 Bwg	1.0
16 Bwg	0.99

3. Relative to 6 rows of tubes, the effect of the number of tube rows on cost is:

Rows	Cost factor
4	1.10
5	1.05
6	1.00
8	0.95

4. Relative to length of 24 ft, the effect of tube length on cost is:

Tube length, ft.	Cost factor
10	1.15
12	1.13
14	1.11
16	1.08
18	1.06
20	1.05
24	1.00
30	0.95
32	0.93
36	0.89
40	0.85

5. Because of shipping limitations the widest tube bundle that can be shop fabricated and shipped to a plantsite is 12 ft. Wider bundles must be field fabricated.

As noted previously, various methods of fan control are used. The primary criteria are temperature control of the process fluid and power consumption. It is feasible to drive *variable speed* fans with standard induction motors using some type of a.c. adjustable frequency drives (AFD) – a variable voltage inverter, a pulse width modulator or current source types.

One alternative is to use a *variable pitch* fan. It offers rather precise temperature control, provides energy savings and is convenient for cold weather operations.^(13,44) It also tends to cost more and may involve more routine maintenance. The choice between variable speed and variable pitch depends on local circumstances and the biases of the purchaser. If power costs are large and temperature control is critical, one or the other normally will be chosen.

The other control alternatives are fluid by-pass, on-off operation (with possibly several fans per cooling bay) and the use of louvers or shutters. By-pass and louvers may be effective in some cases but they are energy inefficient. On-off fan control is simple and may be used if there are a lot of fans in the same service. Winter protection is required in cold climates. In this case, the use of louvers plus some form of variable air rate control is desirable.^(13,42) This is one case where a variable pitch fan plus louvers may be the best system to control internal air circulation.

Outlet temperature is controlled primarily by air rate. Louvers, variable pitch fan blades, and variable speed motors are all used to control temperature. Louvers may be manually adjustable for seasonal or night-day air temperature changes, or controlled automatically. We have found automatic louver control less than satisfactory in those cases where a close tolerance is required on outlet fluid temperatures and the louvers are operating almost closed (where a small change in position causes a large change in air flow rate). In those cases where large air temperature changes are encountered, a variable pitch fan may prove efficient. Some report trouble with the pitch control, but this has not been a problem in my experience. Pitch and speed controls are expensive but in this era of high energy costs they can prove profitable.

Fan power is an important operating cost consideration. A ten percent change in air flow rate will cause about a 35 percent change in power used, assuming efficiency stays constant. Actual power consumption required for a given heat transfer depends on many factors. One is the clearance between the fan and the fan ring. Close clearances are more expensive to fabricate. Consider this in comparing capital cost from different vendors.

There are several rules-of-thumb used by some for fan specification. One is that the distance between tubes and fan should be 0.4-0.5 of fan diameter to allow good air mixing. Another is that the ratio of fan ring area to tube area should not be less than 0.4.

The total cost of air cooling is very sensitive to the temperature approach between entering air and the cooled fluid leaving. It is seldom that an approach less than 10-11°C [18-20°F] can be justified. Most coolers use a 22-28°C [40-50°F] approach to ambient air temperature. Anything larger than this is clearly uneconomical.

The total cost (operating plus amortized capital cost) increases sharply as approach decreases. An approach of 11°C [20°F] has a total cost about 1.6 times that of a 22°C [40°F] approach.

This approach limitation adversely affects the economics of aerial cooling in many cases. Using a 38°C [100°F] D.B. and a 24°C [75°F] W.B. affords a reasonable comparison between air and water cooling. The fluid leaving an air cooler with a 16°C [29°F] approach will be at 54°C [129°F]. If a cooling tower exchanger is used, the temperature leaving likely will be about 42°C [108°F]. This 12°C [22°F] difference can have a significant effect on the size (and cost) of equipment affected by the temperature level achieved in the coolers.

Operators usually prefer air coolers, but they sometimes pay a high price for the convenience. Examine the alternatives carefully before making a choice.

Offshore installations in temperate or cold climates can use both sea water and aerial cooling effectively. On many platforms both occur. How and when each is used depends on temperature level, cooling load, space availability, etc.

COMBINATION COOLERS

Figure 13.24 shows one type of combination cooler that may offer advantage where water is expensive to treat or is in short supply. It is an aerial cooler preceded by an evaporative section. The entering air is cooled with water as necessary. When the air temperature is low enough the water may be shut off. Water rate may be decreased at intermediate temperatures.

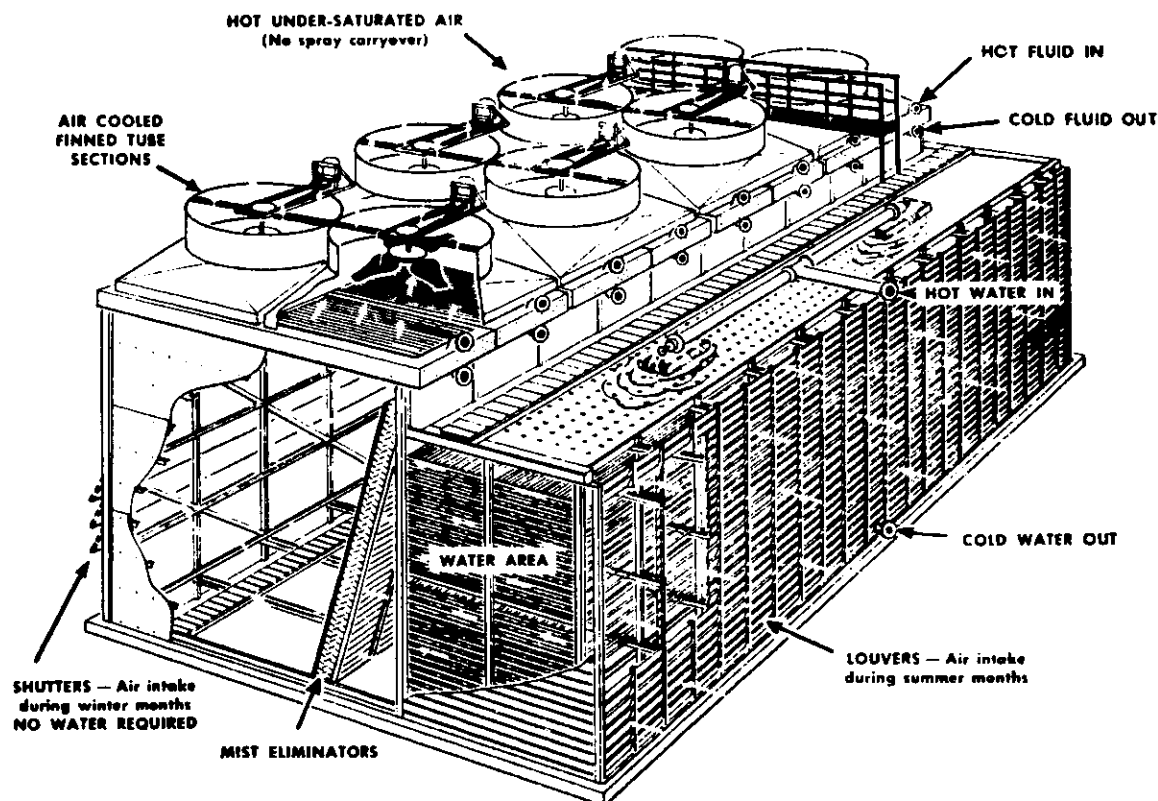


Figure 13.24 Combination Air-Water Cooler

This unit has a higher capital cost but may offer a total cost saving in some applications. When equipped with controls to vary fan horsepower output this unit offers flexibility at minimum operating cost.

Some have accomplished a similar result by using an air cooler for water. This cooling is supplemented as needed with a cooling tower. The combination unit is usually preferred.

CHOICE OF HEAT EXCHANGERS

It should be apparent from preceding discussions that choice of heat exchangers involves many factors. It is relatively easy to choose one that will work. An intelligent choice involves choosing equipment that optimizes cost of the total system without compromising operating reliability.

Heat exchangers normally cost less per unit of energy transferred than any other type of energy equipment. If you "chisel" on exchanger size you must pay dearly for this in the cost of companion equipment in many instances. Since heat loads vary with flow rates, some flexibility must be provided. If done wisely, a little extra heat exchange capacity is the cheapest "insurance" one can purchase.

There are some "rules" one should follow.

1. Do not specify or purchase a HEX without consideration of its effect on the total process.
2. Do not make the capital cost of the HEX alone a sole criterion for purchase.
3. Acquaint the vendor with details of service and point out the choice will be made on both initial and operating cost, not initial capital cost alone.
4. Use realistic pressure drop specifications since this affects size and cost. Allow as much pressure loss as economics dictates for the actual system and not merely reproduce a standard spec that might not apply.

Remember: the vendor presumably understands his product but he knows only as much about the application of his product as the customer conveys to him. The majority of less-than-satisfactory exchanger installations is as much the fault of the customer as it is the vendor.

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NOTES:

14

PUMPS

The choice of a pump is not easy. There are thousands of manufacturers of the dozen or so basic models of pumps. Within each model there are numerous combinations of metallurgy, bearings and seals, packing, mechanical configurations, etc. The final choice involves input from one experienced in pump applications and a careful appraisal of pump needs.

However, there are some basic principles and criteria that apply to all pumps. These will be reviewed in this chapter. The emphasis is on process type pumps used in surface facilities although a brief overview is provided for downhole (wellbore) production pumps. The Standards of the Hydraulic Institute^(14.1) are the basis of the specification of process pumps. Downhole pump standards are set forth by the American Petroleum Institute and individual companies.

Basic Process Pumps

These may be divided into two categories: *positive displacement* and *kinetic* or *dynamic*. The former increase pressure by direct mechanical action. Kinetic pumps impart kinetic energy to the fluid, a portion of which is converted to pressure by reduction in velocity. It is convenient to subdivide these categories.

Positive Displacement

A. Reciprocating

1. Piston
2. Plunger

B. Rotary

1. Gear
2. Screw
3. Vane

Kinetic (or Dynamic)

A. Centrifugal

1. Radial flow
2. Axial flow
3. Mixed flow
4. Special high head, low flow

B. Peripheral (regenerative turbine)

The choice of type depends on liquid density and viscosity, the presence of solids or entrained gases, liquid corrosivity, head (pressure differential across pump) and liquid flow rate. These factors are discussed in detail in many books and trade journal articles.

Viscosity can have a profound effect on performance, particularly of centrifugal pumps. In pump literature you may encounter the viscosity unit SSU (Saybolt Seconds Universal). SSU is the time in seconds that it takes 60 cm³ of liquid to flow out of a standard Saybolt viscosimeter. The conversion formula is

For $SSU < 100$, μ , in cp = $\gamma [(0.226 SSU) - (195/SSU)]$

For $SSU > 100$, μ , in cp = $\gamma [(0.220 SSU) - (130/SSU)]$

Almost all of the crude oils, condensates and process fluids in the petroleum industry possess a viscosity less than 500 SSU, a value below which viscosity has little effect on pump choice. Above this viscosity, a positive displacement pump probably is the best choice.

Solid-liquid slurries present a particular problem for all rotating pumps. If used at all in this service, slower speeds must be used. When using a pump for reservoir liquids, remember that consideration must be given to the possibility that solids also may be produced. A special "slurry" pump may be necessary if solids separation is not provided, particularly early in well life.

If the liquid contains over 4-5% entrained gas, the use of a positive displacement pump is indicated.

A corrosive liquid usually requires special linings or metallurgy. Low velocity normally is desired, since erosion is the natural companion of corrosion.

Subject to the above limitations, pump type chosen usually depends to some degree on the combination head and liquid flow rate considerations. Figure 14.1 shows the typical range of performance of common pump types.

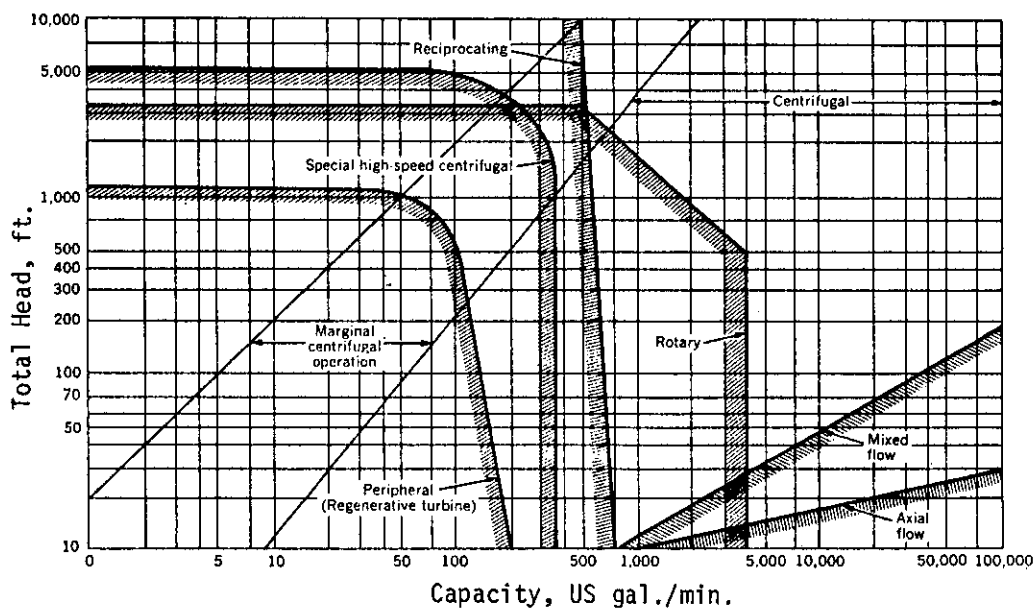


Figure 14.1 Typical Application Range of Common Pumps^(14.2)

Units for Head and Flow Rate

The relationship between head and ΔP is represented by the equation

$$\Delta P = A \gamma H \quad (14.1)$$

Where: ΔP = outlet-inlet pressure
 γ = liquid specific gravity
 H = head across pump
 A = unit conversion factor

Metric	English
kPa	psi
m	ft
9.81	0.433

The head in feet is 3.281 times the head in meters.

In America it has been traditional to use "gpm" as an abbreviation for U.S. gallons per minute. The use of cubic feet or cubic meters, per hour or per second, is sometimes more convenient for calculation purposes. The following conversion factors may prove useful:

$$1 \text{ m}^3 = 35.31 \text{ ft}^3 = 264 \text{ U.S. gal} = 220 \text{ U.K. gal}$$

GENERAL CHARACTERIZATION

There are two calculations made for all types of pumps that are used for specification and choice purposes.

Net Positive Suction Head (NPSH)

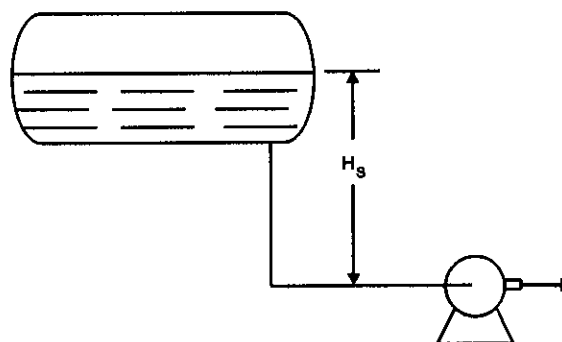
NPSH is a critical factor in pump selection. It is the positive head required for proper head performance.

Sufficient NPSH must be provided to prevent formation of small gas bubbles which then collapse, releasing energy that can damage the pump. This is called *cavitation*.

From a phase behavior viewpoint, said bubbles can only occur when the pressure is at or below the bubblepoint. So, the pressure at the suction flange must be enough above this bubblepoint to prevent gas from forming.

NPSH in a centrifugal is a function of pump design and is governed by rotation speed (N), inlet (eye) area of the impeller, type and number of vanes, and like factors. With reciprocating pumps, NPSH depends on speed and valve design. Involved is what is called acceleration head, caused by the reciprocating motion.

In pump selection one must provide an available NPSH that is equal to the NPSH required by the manufacturer's specifications. If possible, this is done by elevating the suction vessel above the pump, as shown on the right.



The NPSHA available for pumps may be found by the following:

$$\text{NPSHA} = \frac{A (P_s - P_v - \Delta P_f)}{\gamma} + H_s - H_m - \frac{v^2}{2 g_c} - H_{ac} \quad (14.2)$$

Where:

- P_s = absolute pressure of suction vessel
- ΔP_f = fitting and friction loss in suction line
- H_s = height between lowest drawdown line of suction vessel and centerline of pump suction
- H_{ac} = acceleration head (for reciprocating pumps only)
- v = velocity at pump entry
- H_m = extra available head (margin) above minimum required as determined by company or personal specification
- P_v = vapor pressure of liquid at pump

Metric	English
kPa	psia
kPa	psi
m	ft
m	ft
m/s	ft/sec
m	ft
kPa	psia

The value of A will depend on the units used for H and P, as shown in the table below.

P	H	A	NPSHA
bar	m	10.22	m
kPa	m	0.102	m
psi	ft	2.31	ft

Generally, $P_s = P_v$ because of equilibrium in a closed tank. In this case, line size and line length directly affect the ΔP_f term. Increasing line size decreases pressure loss.

The value of NPSHA must equal or exceed the manufacturer's NPSHR. Increasing the pipe size sometimes may not be convenient. One may raise the suction vessel, lower the pump, use a vertical "deep well" pump, install an inducer, or replace the pump with one having a lower NPSH requirement.

The margin, H_m , (usually a company or personal specification) is advisable because several factors influence pump suction design other than NPSH. Vortexing, short radius elbows, "belly-up" pipe reducers, long suction lines and numerous fittings should be avoided.

The ΔP_f term is calculated by the usual equations. The length, L, must be increased by the equivalent length of all fittings placed on the suction side. This length addition may be as high as three times the actual length of suction pipe!

The velocity head ($v^2/2g_c$) increases from near zero in the vessel to some finite value at a centrifugal pump suction and such energy must be extracted from the original static head. In most instances, this value can be neglected. Larger pipe sizes, of course, yield lower velocity heads and lower losses.

Reciprocating pumps require a correction called *acceleration head* (H_{ac}). When designing reciprocating pump inlet piping with no pneumatic pulsation control device, one must provide for the continual acceleration and deceleration the fluid undergoes with plunger movement. The following equation reasonably defines the acceleration head for suction pipes less than 15 m [50 feet].

$$H_{ac} = \frac{A C K N L q}{d^2} \quad (14.3)$$

Where:

- H_{ac} = acceleration head
- N = rotational velocity of crankshaft
- L = length of suction pipe
- q = pump rate
- d = suction pipe diameter
- A = coefficient
- C = fluid velocity factor:

- 0.0158 for single acting, simplex power pump
- 0.0079 for double acting, simplex power pump
- 0.0079 for single acting, duplex power pump
- 0.0046 for double acting, duplex power pump
- 0.0026 for single acting, triplex power pump
- 0.0016 for single acting, quintuplex power pump
- 0.0011 for single acting, septuplex power pump

- K = fluid compressibility factor:
- 0.4 compressible liquids (C_2 , etc.)
- 0.5 most hydrocarbons
- 0.67 amine, glycol, water
- 0.71 deaerated, hot water

Metric	English
m	ft
rev/min	rev/min
m	ft
m^3/s	ft^3/sec
m	ft
3.32	1.0

The value determined must be subtracted from the NPSHA already calculated and still maintain a reasonable margin of available head. Line size is inversely proportional to acceleration head, i.e., the larger the diameter, the smaller the acceleration head loss.

In suction lines longer than 15 m [50 ft], internal disturbances and pressure wave velocities become significant, and the acceleration head found by the equation probably would be more than actually existed.

The introduction of properly-sized suction side pneumatic or gas pulsation bottles adjacent to the pump practically eliminates acceleration head. Installation of these reduces the size of suction pipe significantly.

One fitting usually forgotten in design is the startup screen normally placed in the inlet line. Destructive cavitation may occur during startup caused by this oversight, so include it in the list of fittings. Unfortunately, published data is limited, but a judgmental allowance should be made.

Pump vendors seldom take such inlet screens into consideration for their NPSHR values. Some specifically prohibit their use after startup. To be safe, question the vendor about screen use with his equipment.

Pump suction, then, should be examined carefully, using line sizes for pipe and fittings which provide adequate suction head margin at the pump, during and after startup procedures.

Specific Speed

This is a type of index that can be applied to all types of pumps to aid in their selection and use. The equation is:

$$N_s = \frac{(A)(N)(q)^{0.5}}{H^{0.75}} \quad (14.4)$$

Where:

A	=	conversion constant
N	=	pump speed
q	=	liquid flow rate
H	=	head

Metric	English
1.0	0.0194
rpm	rpm
m ³ /s	gpm (U.S.)
m	ft

The primary use of specific speed is in the classification of centrifugal pumps.

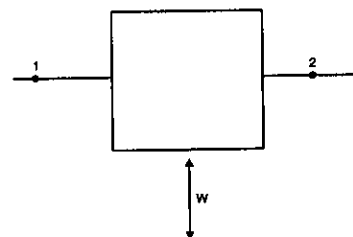
PUMP POWER REQUIREMENTS

The amount of power required to pump a given amount of liquid to the desired pressure is an application of the First and Second Laws of Thermodynamics discussed in Chapter 7 (Volume 1). The procedure is the same for all pumps.

The basic approach involves establishing the energy gained by the fluid involved, assuming that this energy change is accomplished adiabatically and reversibly (isentropically). This theoretical or reversible energy is corrected to an actual energy by means of a thermodynamic efficiency term which has been established by test. This efficiency encompasses the error in the isentropic assumption as well as true mechanical energy losses in the machinery.

The energy balance around the equipment may be reduced to the form

$$\int_{P_1}^{P_2} V dP = \Delta H = W_{\text{theor}} = W_{\text{rev}} \quad (14.5)$$



The subscripts "theor" and "rev" mean the same thing - work is done isentropically on the fluid. The work will be plus or minus, depending upon whether it is gained or lost by the fluid.

The actual (shaft) work that must be provided by the driver is found from the equation

$$\begin{array}{cc} \text{Pumps, Compressors} & \text{Expanders} \\ W_{\text{act}} = \frac{W_{\text{theor}}}{E} & W_{\text{act}} = (E)(W_{\text{theor}}) \end{array} \quad (14.6)$$

where E is written as a fraction from zero to one, even though it is often expressed as a percentage.

The use of ΔH to find work theoretical is limited primarily to compressors and gas expanders. Liquid systems use the integral to find work.

The work done by a pump is determined on the assumption that the liquid is incompressible. Since ΔH is small for most pumps, we usually apply Equation 14.5 as follows

$$V(\Delta P) = W_{\text{theor}} \quad (14.7)$$

where "V" can be expressed as a specific volume or as a volume rate of flow.

The theoretical equations use the term "work" but we actually are concerned with power, the time of doing work. So, W is power in the actual case.

The traditional work term for combustion engines has been brake horsepower (BHP), the power transmitted by the engine shaft. For electrical motors, the kilowatt (kW) has been traditional. In the SI metric system the watt (W), with an appropriate prefix, is the standard power unit. The term "horsepower" should gradually disappear. This is a trivial problem since

$$1 \text{ horsepower} = 0.746 \text{ kW}$$

Equation 14.7 may be expressed in working form as

$$kW = \frac{A q (P_2 - P_1)}{E} \quad (14.8)$$

Where: A = constant
q = volume rate of liquid flow
P₂ = outlet pressure of pump
P₁ = inlet pressure of pump
E = thermodynamic (overall) efficiency

Metric	English
1.0	4.35×10^{-4}
m ³ /s	gpm (U.S.)
kPa	psia
kPa	psia
-	-

Note: In some case where flow rate is small, q may be expressed in liters per second (L/s). In this case, A = 0.001 in Equation 14.8, when all terms are in metric units.

The value of q may be found from one of the equations

$$q = m v = \frac{m}{\rho}$$

Where: m = mass rate/unit time v = specific volume ρ = density

The value of "E" will depend on the pump type and the service. The values vary widely, particularly for centrifugal pumps.

The value of ΔP from Equation 14.1 may be substituted for (P₂ - P₁) in Equation 14.8 to calculate power directly from fluid head.

CENTRIFUGAL PUMPS

Part (a) of Figure 14.2 shows a schematic of a single stage centrifugal pump. The liquid enters the "eye" at the left, then into the wheel or impeller in the middle, is ejected at high velocity at the circumference, and leaves at the top outlet flange. The amount of head produced depends on the velocity imparted by the wheel.

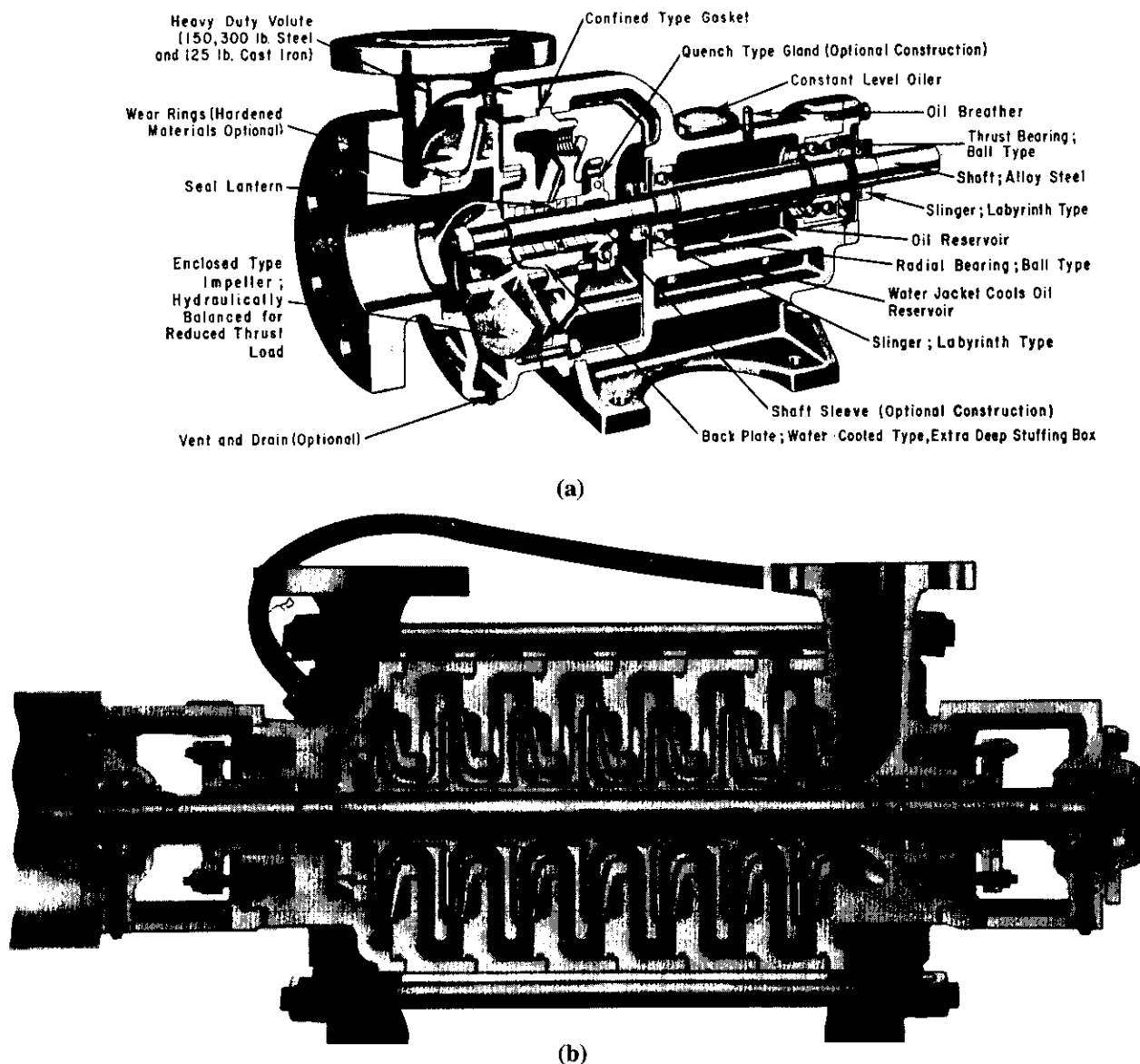


Figure 14.2 Cutaway of Centrifugal Pumps

This is a closed impeller pump with wear rings, which limits the amount of gas that can leak from the high pressure side to the low pressure side. The result is an increase in the efficiency and head achieved.

In many cases, the head produced by a single stage pump is less than that required. A large number of wheels can be placed in series on the same shaft to achieve multi-stage performance. This is shown in the

lower portion of Figure 14.2. Note that the discharge from a stage is the inlet to the next one. Whether the pump is horizontal (as shown) or vertical, the number of stages is limited only by temperature and mechanical factors. A view of a multi-stage, vertical centrifugal pump is shown later in Figure 14.12.

A number of wheel types, sizes and mechanical configurations are used in centrifugal pumps. Each pump has a *characteristic curve*, of the types shown in Figure 14.3, for two different single stage pumps. Notice that both are similar in nature but that specific values vary widely.

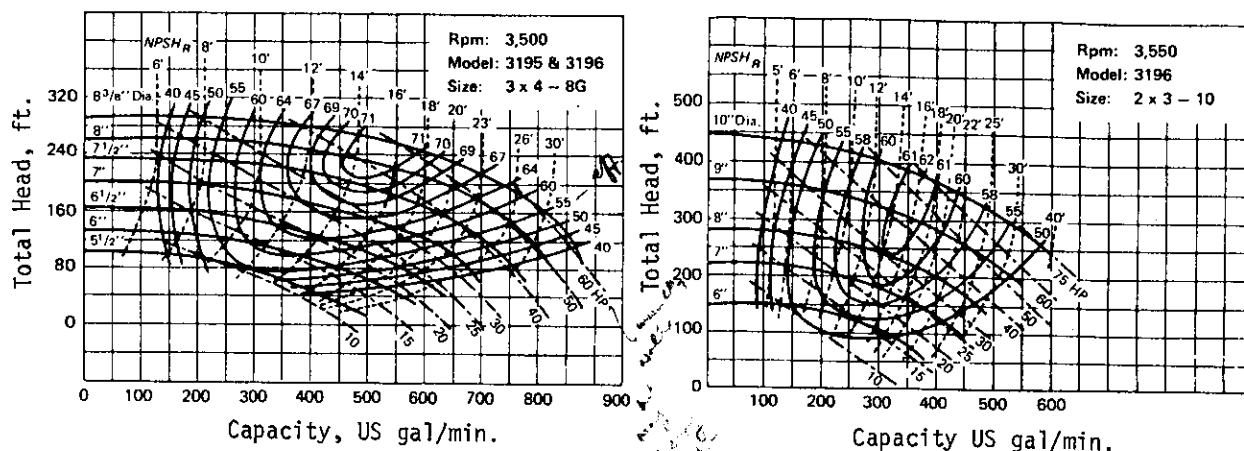


Figure 14.3 Typical Characteristic Curves for Centrifugal Pumps

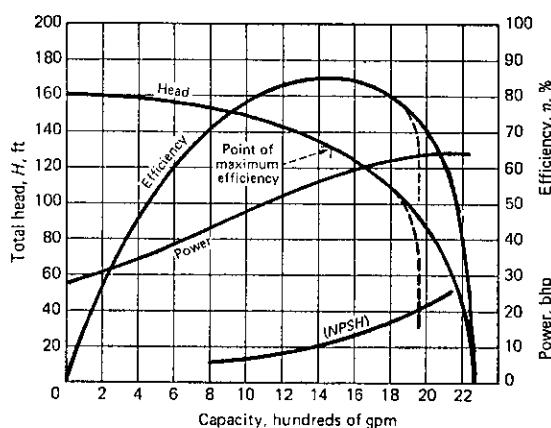


Figure 14.4 Performance Characteristics for a Centrifugal Pump^(14.3)

Shown as a function of head, flow rate and impeller diameter are NPSHR, power and overall efficiency. From information like this, one can choose an appropriate pump. The final choice will involve a number of practical considerations, including economics.

Notice how the efficiency lines curve and reverse themselves. For a given diameter wheel there are two combinations of head and rate that possess the same efficiency.

Once a pump has been selected, and the impeller diameter fixed, the performance curves can be plotted as shown in Figure 14.4. The characteristic head, efficiency, power and NPSH curves depend on the impeller type. Figure 14.4 depicts a characteristic curve for a typical process pump with $N_s \approx 40$.

Specific Diameter

Balje^(14.4) has proposed a concept, specific diameter, which can be used with specific speed to make a general choice of pump type. Specific diameter, d_s , is defined by the equation

$$d_s = \frac{(A)(d)(H)^{0.25}}{(q)^{0.5}} \quad (14.9)$$

Where: d = impeller diameter
 q = liquid flow rate
 H = head
 A = conversion constant

Metric	English
m	ft
m^3/s	gpm
m	ft
0.743	22.15

Figure 14.5 shows the relationship between d_s and N_s for a single stage centrifugal pump.

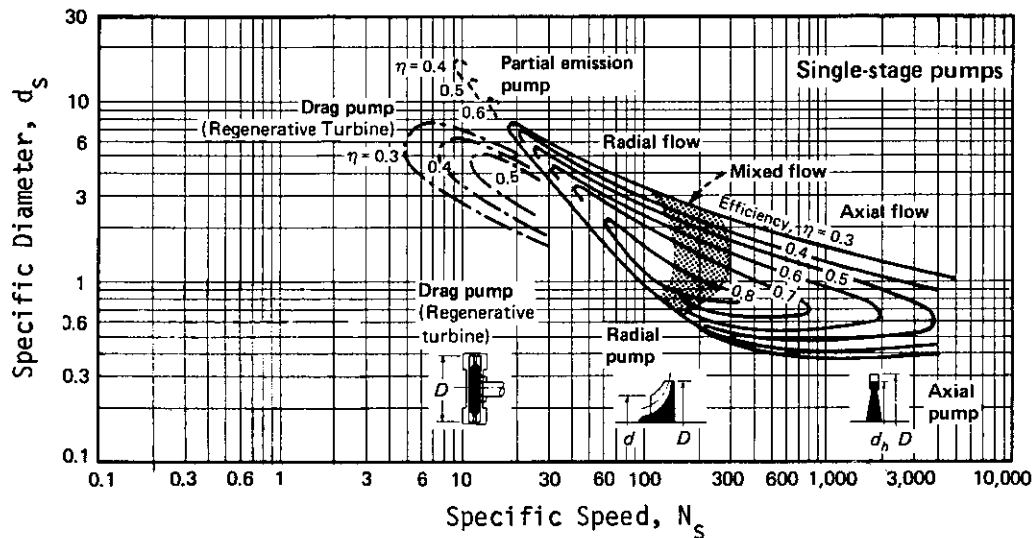


Figure 14.5 Specific Speed vs. Specific Diameter for Single-Stage Centrifugal Pumps

The conversion constants in Equation 14.9 above are those required to express d_s in the units used in Figure 14.4. Likewise the N_s values from Equation 14.4 must be multiplied by 2.44 before using Figure 14.5. Shown is the effect of specific diameter and speed on efficiency and the general type of pump used. Figure 14.5 can be used to limit the types of centrifugal devices chosen for more detailed consideration. A final choice cannot be made without a detailed characteristic curve.

Figure 14.6 shows the effect of specific speed on the likely maximum efficiency of centrifugal pumps. Also shown is the impeller characteristic consistent with a particular specific speed. Each curve is for a similar family of pumps with each point being a different pump. It is apparent that peak efficiency occurs for values of N_s between 30 and 60 in Figure 14.6. The use of multiple stages is indicated when too high a head has an adverse effect on specific speed, hence efficiency.

Suction Specific Speed

Cavitation can occur even though NPSHA exceeds NPSHR. This has led to the development of the *suction specific speed* concept.^(14.5)

$$N_{ss} = \frac{A N q^{0.5}}{(\text{NPSH})^{0.75}} \quad (14.10)$$

Where: N = rpm
 q = liquid flow rate
 NPSH = required NPSH
 A = unit conversion factor

Metric	English
—	—
m^3/s	gpm
m	ft
1.0	0.0194

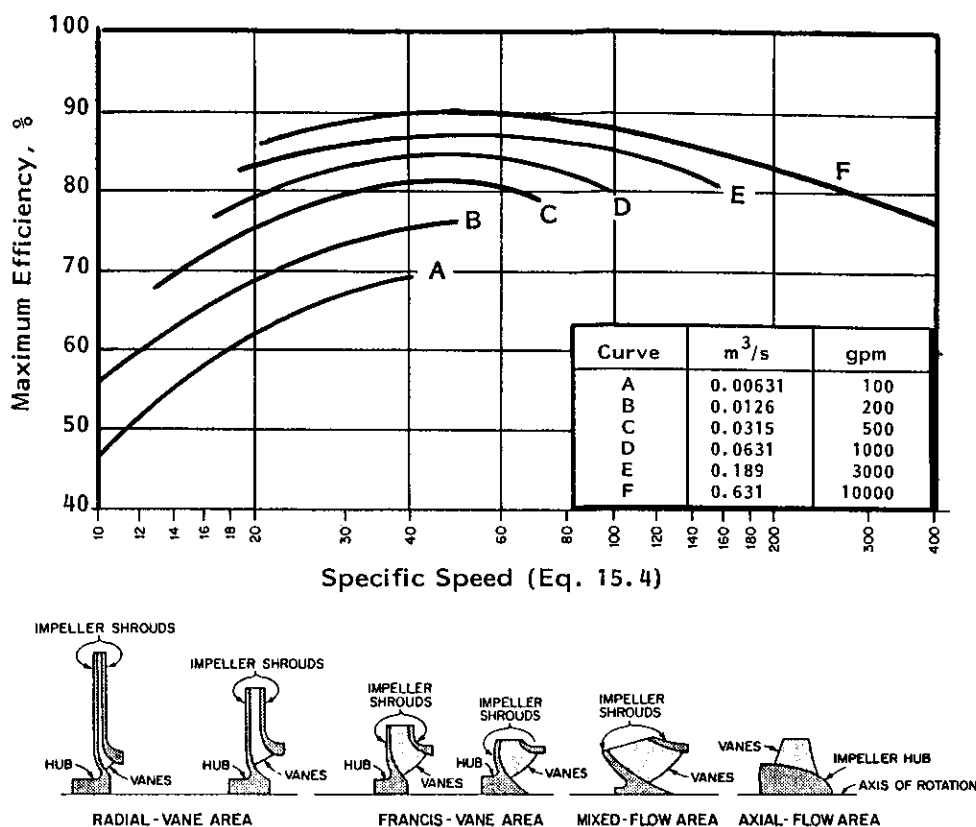


Figure 14.6 Overall Efficiency vs. Specific Speed for Centrifugal Pumps^(14.5)

Cavitation may occur at low flow conditions if N_{ss} is greater than about 120. It has been shown that in water service, cavitation can occur at 30-40% of best efficiency flow even though the NPSHA is 2.0 to 2.5 times greater than the NPSHR from the manufacturer's curves. This is due to internal recirculation in the impeller. In hydrocarbon service this phenomena is less troublesome, but in any case it is recommended that pumps with an N_{ss} greater than 200 be avoided. Once a pump has been chosen, it is advisable to make this calculation, particularly for pumps expected to operate at well below their design rate for part of the time. Pump bypasses are recommended for high energy centrifugal pumps. Reference 14.7 provides an excellent summary of pumping conditions which warrant bypass installation.

Pump Selection and System Performance

Selection of a centrifugal pump requires knowledge of the hydraulic characteristics of system in which the pump is to be installed. The "system head" is the amount of energy which must be imparted to the liquid at the pump to achieve the required flow rate. System head is made up of the two components static head and dynamic head. Static head arises due to a difference in elevation or pressure level. It is independent of system flow. Examples of static head include the elevation difference between sea level and platform deck in a sea water supply pump or the difference between the surge tank and contactor pressure in an amine circulation pump. Dynamic head is due to friction losses in the system. It is the ΔP_f term (expressed in head) discussed in Chapter 10.

Figure 14.7 shows a system head curve and a typical pump head curve. The intersection point of the two head curves represents the flow rate which will be delivered by the pump at a particular speed.

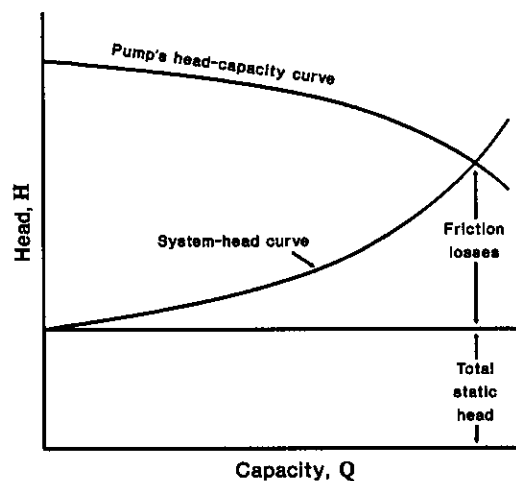


Figure 14.7 System Requirements vs. Pump Capacity

Control

In process applications it is unusual for a system to require operation at a single fixed capacity. The system demand is frequently variable. Remember that a given pump operating in a system will deliver the flow rate which corresponds to the intersection of the pump head and system head curves. In order to control the flow through the pump it is necessary to change the shape of either one or both of the curves.

The traditional method for controlling the flow through any dynamic pump like a centrifugal has been to use a valve on the outlet. Liquid is throttled through this valve so that the system head is artificially increased to give the desired flow in accordance with the pump's characteristic curves. This has been a satisfactory arrangement, even though it possesses some basic mechanical faults. From an economic viewpoint, the primary fault is that it is inefficient from an energy consumption viewpoint. For this reason (among several), variable speed drive systems are being employed. With a variable speed drive the pump head curve is adjusted in accordance with the affinity laws (Equation 14.11). Figure 14.8 illustrates the use of these two control strategies.

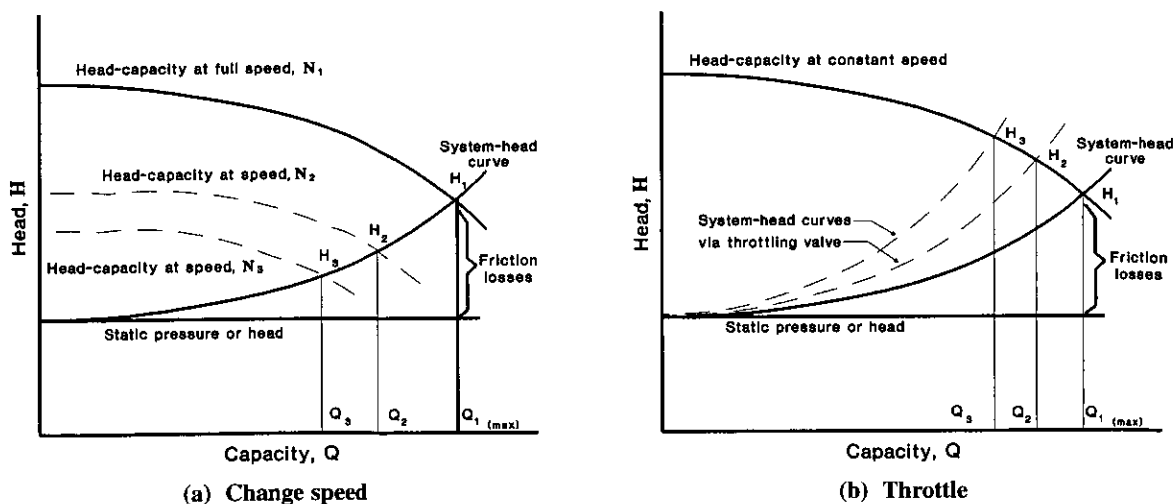


Figure 14.8 Capacity Control of a Centrifugal Pump^(14.3)

Considerable power savings can be realized in some systems through the use of variable speed drivers. Figure 14.9 illustrates the magnitude of these savings in one system.

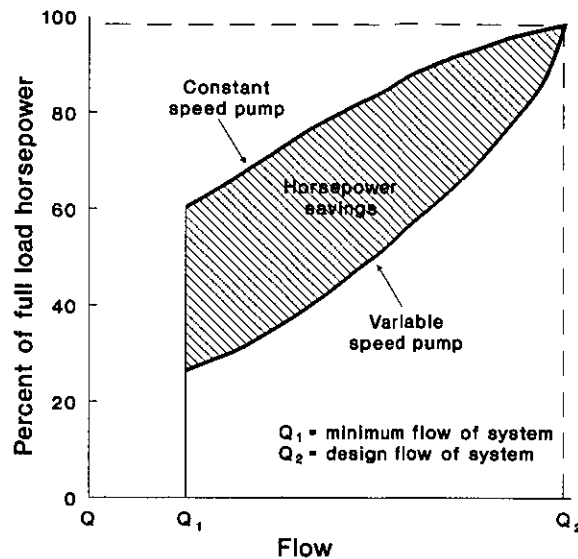


Figure 14.9 Potential Power Savings from Variable Speed Drives

There also are some other advantages of variable speed systems. Soft startups are possible without drawing a large amount of current, since there is no closed discharge valve. Pump wear is reduced. It never has to operate with artificially high case pressures. At constant speed, shaft deflection usually increases with decreasing flow. With variable speed, said deflection is less. Hydraulic noise also is decreased. Last, but not necessarily least, throttling valve maintenance is eliminated.

There are five basic methods for varying pump speed: solid state a.c. and d.c. motor control, mechanical, electro-mechanical and fluid speed changers between the driver and pump. Discussions of these abound in the process literature.^(14.8, 14.9)

Multiple Pumps in Series

In some cases the total head required might be obtained from several pumps in series. If this is done, the pumps must be matched so that they operate harmoniously together. In doing so, one must recognize what is usually called the *affinity law*. This law is represented by the equation

$$\frac{N_1}{N_2} = \frac{q_1}{q_2} = \frac{(H_1)^{0.5}}{(H_2)^{0.5}} = \frac{(kW_1)^{1/3}}{(kW_2)^{1/3}} \quad (14.11)$$

Where: N is speed, q is flow rate and H is head. Subscripts (1) and (2) refer to two different conditions for the same pump or could be applied to two different pumps, to obtain matching performance.

Equation 14.11 is for a fixed impeller diameter. If rpm (N) remains constant and impeller diameter is changed, multiply

1. Original capacity by (d_2/d_1)
2. Original head by $(d_2/d_1)^2$
3. Original kW by $(d_2/d_1)^3$

This can be used to convert performance expected for an individual pump from a known condition to a new condition. It also can be used as a part of the pump matching process.

General

Centrifugal pumps are very flexible and offer a high power output per unit weight and volume. However, like all high speed devices, they can offer unsatisfactory reliability if proper attention is not given to a host of mechanical details. Table 14.1 lists 89 sources of problems. In perusing this list, one can recognize the mechanical considerations that need attention in pump selection and specification.

TABLE 14.1
"Checklist" of Centrifugal Pump Problems ^(14.10)

1. Measuring instruments not correctly calibrated and/or incorrectly mounted.	29. Total head of system either higher or lower than expected.	61. Stuffing in box eccentric in relation to shaft.
2. Air entering the pump during operation, or the pumping system not completely deaerated before starting.	30. Misalignment between pump and driver.	62. Mechanical seal improperly installed.
3. Insufficient speed.	31. Rotating parts rubbing on stationary parts.	63. Incorrect type of mechanical seal for given operating conditions.
4. Wrong direction of rotation.	32. Worn bearings.	64. Internal misalignment of parts, preventing seal washer and seal from mating properly.
5. Discharge pressure required by the system is greater than that for which the pump was designed.	33. Packing improperly installed.	65. Sealing face not perpendicular to shaft axis.
6. Available NPSH too low (including suction lift too high).	34. Incorrect type of packing.	66. Mechanical seal that has been run dry.
7. Excessive amount of vapor entrained in liquid.	35. Mechanical seal exerts excessive pressure on seat.	67. Abrasive solids in liquids that come into contact with seal.
8. Excessive leakage through wearing surfaces.	36. Gland too tight.	68. Leakage under sleeve due to gasket and O-ring failure.
9. Viscosity of liquid higher than viscosity of liquid for which the pump was originally designed.	37. Improper lubrication of bearings.	69. Bearing-housing bores not concentric with water end.
10. Impeller and/or casing partially (or fully) clogged with solid matter.	38. Piping imposing strain on pump.	70. Damaged or cracked bearing housing.
11. Waterways of impeller and/or of casing very rough.	39. Pump running at critical speed.	71. Excessive grease in bearings.
12. Fins, burrs, sharp edges, etc., in the path of the liquid.	40. Rotating elements not balanced.	72. Faulty lubrication system.
13. Impeller damaged.	41. Excessive lateral forces on rotating parts.	73. Improper installation of bearings (damage during assembly, incorrect assembly, wrong type of bearings, etc.).
14. Outer diameter of impeller machined to a lower diameter than specified.	42. Insufficient distance between outer diameter of impeller and volute tongue.	74. Bearings not lubricated.
15. Faulty casting of impeller and/or casing.	43. Faulty shape of volute tongue.	75. Dirt getting into bearings.
16. Impeller incorrectly assembled in casing.	44. Undersize suction and/or discharge piping and fittings (sometimes causing cavitation).	76. Water entering the bearing housing.
17. System requirements too far out on the head/capacity curve.	45. Loose valve or disk in the system, causing premature cavitation in the pump.	77. Balancing holes clogged up.
18. Obstruction in suction and/or discharge piping.	46. Bent shaft.	78. Failure of balancing device.
19. Foot valve jammed or clogged up.	47. Bore of impeller not concentric with its outer diameter and/or not square with its face.	79. Too-high suction pressure.
20. Suction strainer filled with solid matter.	48. Misalignment of parts.	80. Tight fit between line bearing and its seats (which may prevent it from sliding under axial load).
21. Suction strainer covered with fibrous matter.	49. Pump operates at a very low capacity.	81. Pump not primed and allowed to run dry.
22. Incorrect layout of suction and/or discharge piping.	50. Improperly designed baseplate and/or foundation.	82. Vapor or air pockets inside the pump.
23. Incorrect layout of suction sump.	51. Resonance between operating speed of pump and natural frequency of foundation and/or other structural elements of pumping station.	83. Operation at too-low capacity.
24. The operation of one pump (in a system having two or more pumps operating either in parallel or in series, or in a combination of these) seriously affected by the operation of the other pumps.	52. Rotating parts running off-center because of worn bearings or damaged parts.	84. Parallel operation of poorly matched pumps.
25. Water level in suction sump (or tank, or well) below pump intake.	53. Improper installation of bearings.	85. Internal misalignment due to too much pipe strain, poor foundations or improper repair.
26. Speed too high.	54. Damaged bearings.	86. Internal rubbing of rotating parts on stationary parts.
27. Pumped liquid of higher specific gravity than anticipated.	55. Water-seal pipe plugged.	87. Worn bearings.
28. Oversize impeller.	56. Seal cage improperly located in stuffing box, preventing sealing fluid from entering space to form seal.	88. Lack of lubrication.
	57. Shaft or shaft sleeves worn or scored at the packing.	89. Rotating and stationary wearing rings made of identical materials with identical physical properties.
	58. Failure to provide cooling liquid to water-cooled stuffing boxes.	
	59. Excessive clearance at bottom of stuffing box, between shaft and casing.	
	60. Dirt of grit in sealing liquid.	

RECIPROCATING PUMPS

These are positive displacement devices that move an amount of liquid fixed by the piston, plunger or diaphragm displacement on the fluid end. They offer particular advantage with slurries, highly viscous liquids and where a high fluid head is required. Because of the slower speed involved, maintenance costs tend to be less than for centrifugals.

There are two basic types of drives. A *power pump* is driven by a driver like a motor, turbine, etc. A *direct acting pump* is driven by steam, gas, air, etc.

Figure 14.10 shows the fluid end using a plunger and one type of diaphragm. These are the types most commonly employed in petroleum applications. The plunger pump is preferred over the piston pump because the fixed packing provides a better seal than piston rings.

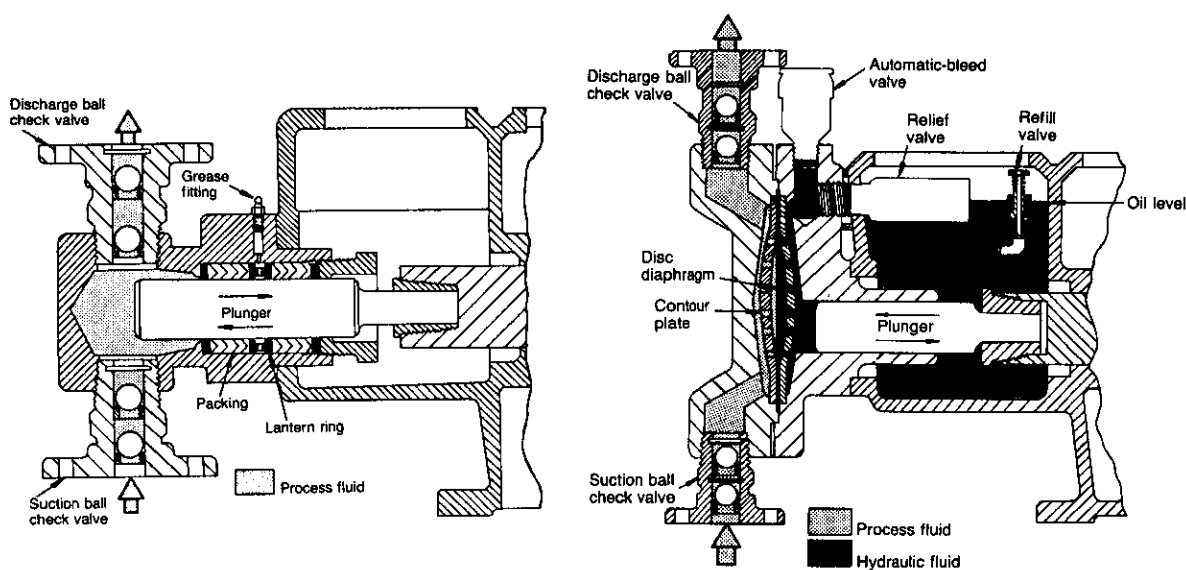


Figure 14.10 View of Fluid End of Reciprocating Pump^(14.11)

The diaphragm usually is employed in corrosive or erosive situations, or where even a small amount of leakage is detrimental. In Figure 14.10, there is a plunger that moves the diaphragm with hydraulic fluid. As it reciprocates it hydraulically moves the diaphragm back-and-forth. Generally, this is preferred over direct coupling of the plunger to the diaphragm.

Ball check valves are shown. These are common in low capacity pumps like metering or chemical feed pumps. Large pumps most often use wing-guided or disk-type valves.

It is common to apply some sort of hard coating to the plunger to minimize scoring and prolong life. Chrome plating, ceramics and nickel alloys are common. In glycol dehydration and sweetening plants (a common application), chrome plating is often used.

The fluid end may be single-acting or double-acting. The pumps shown in Figure 14.10 are single-acting. There is only one inlet and outlet. If a second inlet and outlet are placed on the right side of the plunger, suction will occur at one end while discharge occurs at the other, i.e., the pump is double-acting. In very high head, low flow rate applications the single-acting pump is preferred. In applications where a piston is suitable, a double-acting pump may be preferred.

One basic disadvantage of a reciprocating pump is the velocity surge that occurs during the stroke. One solution is to use more than one pumping element in parallel. A simplex pump possesses only one plunger or piston, a duplex has two, and so on. Shown following are the velocity variations for various pump types, using single-acting pumping elements.

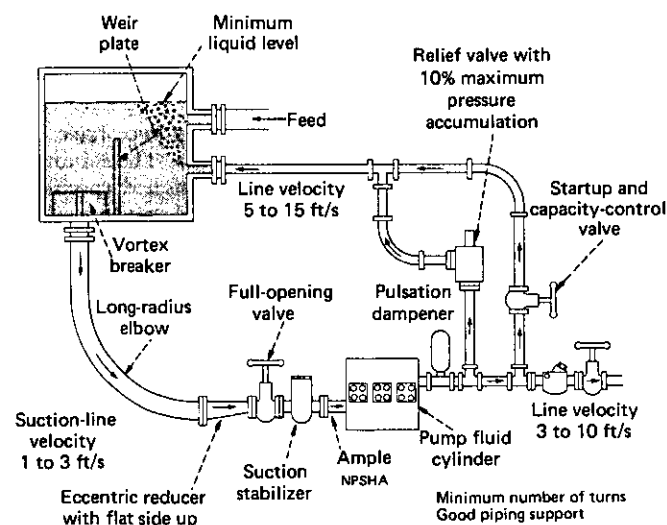
Number	Pump Type	Velocity Variation, %
2	Duplex	160
3	Triplex	25
5	Quintuplex	7
6	Septuplex	4

The velocity variation is the range of velocities related to the average. In a triplex pump the minimum velocity is 82% of the average, the maximum being 107% – a difference of 25%.

A triplex, single-acting reciprocating pump is a common choice in glycol dehydration and liquid sweetening units. It is an effective compromise between pump complexity and surge control.

Plunger diameter will vary from 0.75-2.5 inches with the stroke varying from 1-8 inches. For reliable performance, the allowable pump speed should depend on stroke length, varying from about 1 in. at 600 rpm up to 8 in. at 200 rpm. For a typical 3 in.-stroke triplex pump, a maximum rpm of about 400 rpm is recommended.

The overall efficiency for a powered reciprocating pump typically will be 85-92% when operating at over a 50% load factor. The efficiency of direct-acting pumps will be lower, depending primarily on their speed. The figure following shows the correct hook-up of a reciprocating pump.



Since it is a positive displacement device, a by-pass must be provided for startup and capacity control. Speed and by-passing liquid are the only effective flow control devices available. Reference 14.12, from which this figure is taken, is an excellent review of reciprocating pumps.

ROTARY POSITIVE DISPLACEMENT PUMPS

The rotary pump is a very flexible unit that can handle large quantities of viscous liquid. It is used widely in fluid power applications. Figure 14.11 shows several types of rotary pumps. Of these, the un-timed, three-screw pump is used most commonly in the petroleum industry. It is simple, rugged, has no valves or parts to foul, is relatively quiet and produces pulsation-free liquid.

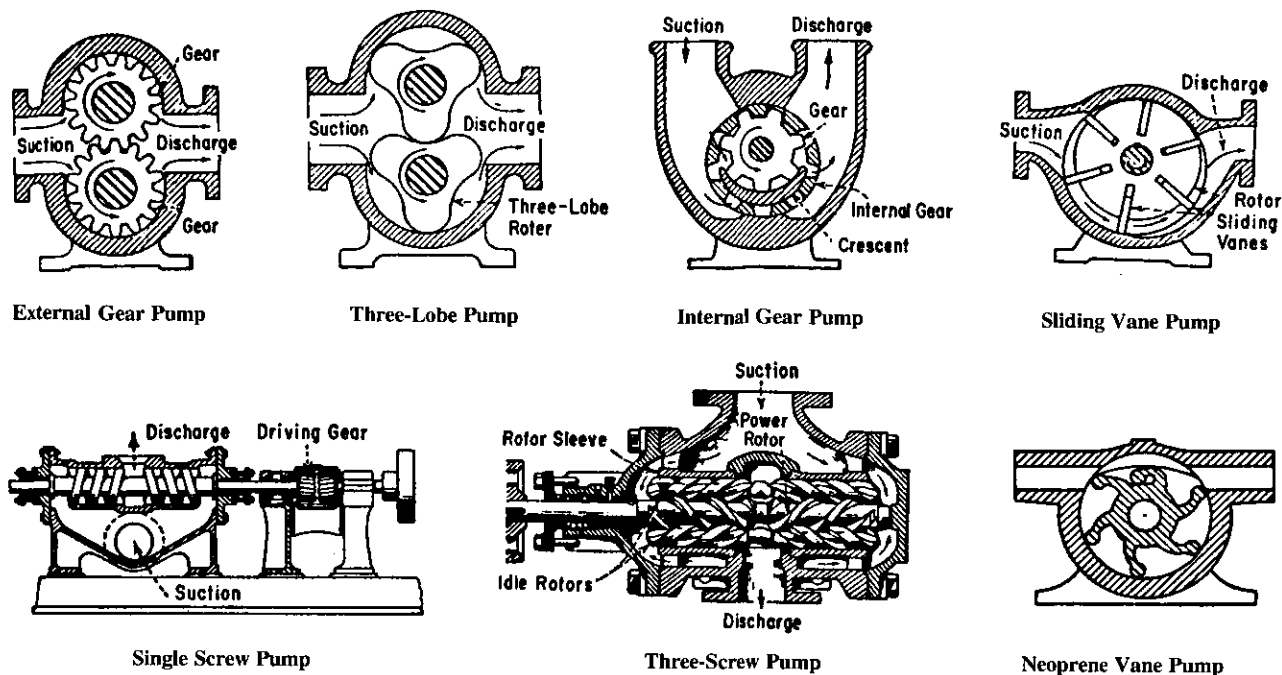
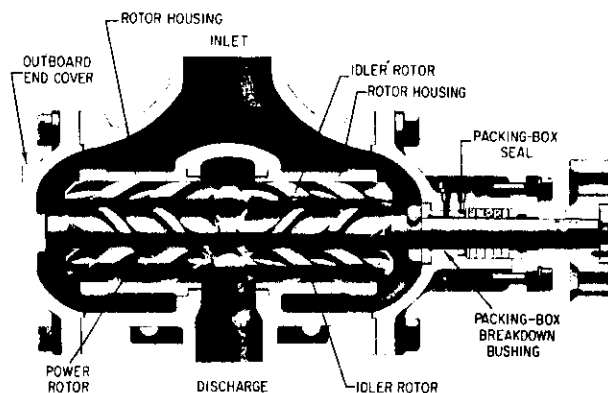


Figure 14.11 Various Types of Common Rotary Pumps

Gear pumps are useful transfer pumps. They may be the best choice when differential pressure exceeds 1030 kPa [150 psi] and viscosity exceeds about 70-80 cp, self-priming is desired and/or entrained vapor compromises centrifugal pump viability. Shown at right is a cutaway view of a double-end gear pump.



Vacuum Pumps

A vacuum pump is really a compressor. It compresses gas, possibly with entrained liquid, from low pressure to atmospheric pressure. The choice depends on service.

The simplest vacuum pump is an ejector whereby a motive fluid is passed through an orifice or nozzle to increase velocity. This produces an artificially low pressure in the "throat" capable of "pumping" gas.

The motive fluid can be any stream on which a pressure drop is feasible and economical. Steam has been used but rising energy and steam costs have reduced the number of applications. In some cases, natural gas is used when a pressure drop is needed anyway. One application is found on glycol regeneration units when a vacuum is needed.

Most vacuum pumps are some version of the vane or lobe type rotary pumps. As with the types shown in Figure 14.11, oil lubrication is required. In service like sea water deoxygenation, some of the water components contaminate the oil and a treating system is required. Thus, "oil ring" pumps offer limited application in this service.

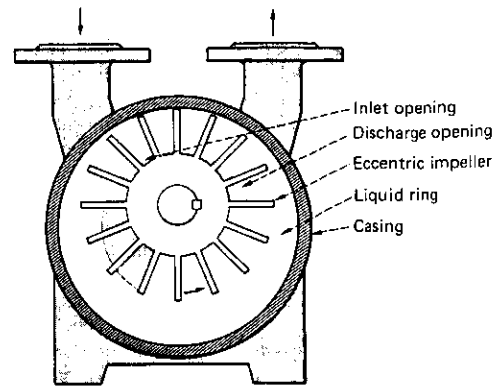
The "water ring" pump may be the choice when pumping vapors incompatible with lube oil. Sea water deoxygenation is one good example of this. A water-ring pump consumes more energy than oil sealed pumps and requires large amounts of seal liquid. With water, this seal is a once-through process.

Roots blowers are dry lobe type pumps whose lubrication system is isolated from process vapors. They will handle large volumes of vapor. In some cases they are operated in series with mechanical pumps to produce the best results. One concern is to keep the compression ratio low enough in the roots blower so that overheating does not occur.

The type of pump depends also on factors like capacity need and the pressure required. In vacuum technology the pressure unit of the *Torr* is commonly used,

$$1 \text{ Torr} = 1 \text{ mm Hg} = 0.133 \text{ kPa}$$

Often it is convenient to express absolute pressure in Torrs instead of in terms of the *P* below atmospheric pressure.



Liquid-Ring Pump

VERTICAL LIFT PUMPS

There are many applications where liquid is to be pumped from a sump, a body of water, a basin, etc. A common pump used is a vertical, multi-stage centrifugal of the type represented by Figure 14.12. The number of stages will vary with the service. A pump much like that shown usually is used as a water "winning" pump in offshore production systems. It can be located at an appropriate depth below the sea water surface to reduce surface debris. Vertical pumps are frequently used in process applications where space is limited. Limited NPSH is available or high head requirements exist.

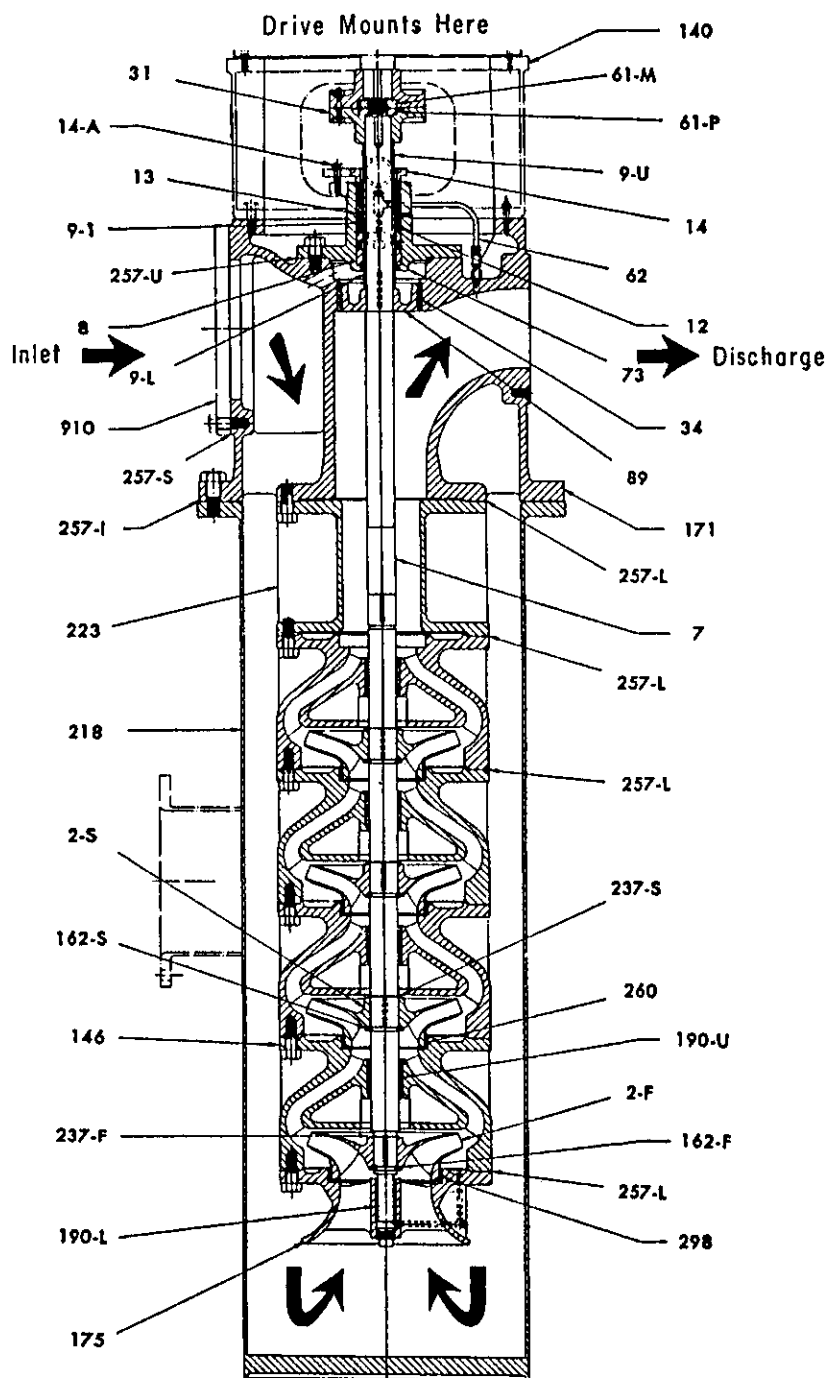
Four basic systems are used to lift liquids in a wellbore. One of these, gas-lift, is outside the scope of this book. The other three are shown in Figure 14.13. Part (a) shows five different rod pumps. All are similar in operation. The upper portion moves back-and-forth. By using two ball valves, the pump barrel is alternately filled and emptied. The amount of liquid pumped is regulated by barrel size, stroke and speed. The pump shown is connected to the surface by "sucker rods."

Part (b) of Figure 14.13 shows a multi-stage centrifugal pump small enough in diameter to be lowered in a wellbore. The basic operating characteristics are the same as regular centrifugal pumps. Most pumps in this service are in the mixed flow range wherein the impeller imparts both centrifugal and axial force on the liquid. Details about this type of pump are found in Reference 14.13.

Part (c) is a hydraulic system. Power oil under pressure is used to drive the pump. Generally, power oil is pumped down the inner pipe string and the mixture of power oil and liquid production return together in the tubing annulus. These are separated and a portion of the oil is used for power oil, to complete the cycle.

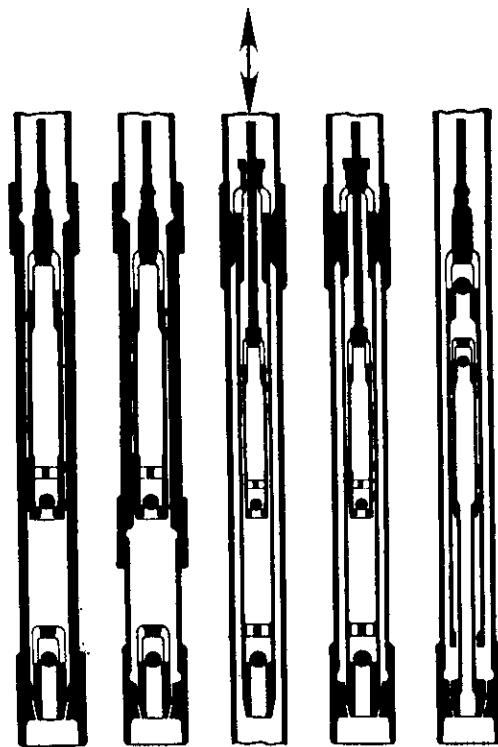
The net pressure on the pump is the column pressure minus the formation pressure; this also is equal to the differential pressure on the engine when the engine and pump pistons have the same diameter.

Each of these basic pump types has an optimum operating range but the final choice often is based on economics. Figure 14.14 shows a summary of relative cost versus pump depth and liquid production

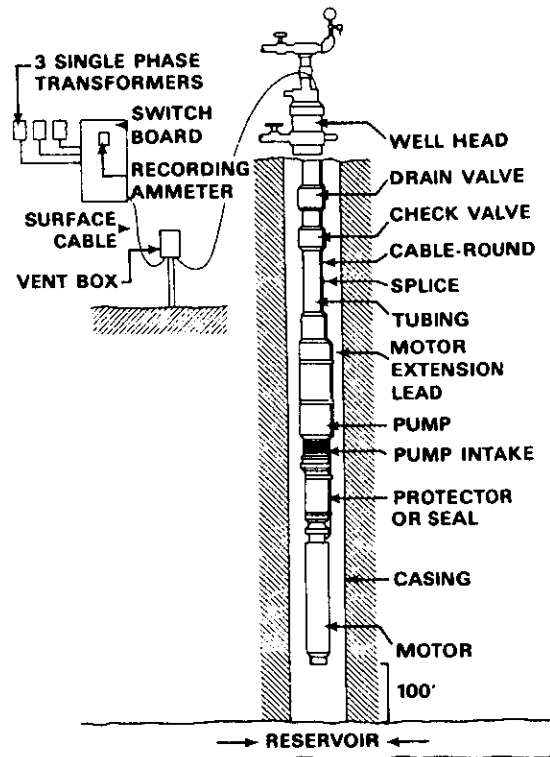


No.	Part Name
2-F	First Stage Impeller
2-S	Second Stage impeller and Above
34	Nozzle Head Bushing
260	Diffuser Ring
298	Suction Bell Ring
8	Stuffing Box Bushing
9-L	Lower Shaft Sleeve
9-I	Intermediate Shaft Sleeve
9-U	Upper Shaft Sleeve
12	Packing
13	Seal Cage
14	Gland
14-A	Gland Bolt
31	Complete Coupling
61-P	Pump Half Coupling Lock Nut
61-M	Motor Half Coupling Lock Nut
62	Complete Piping
73	Stuffing Box
89	Balance Disk
140	Motor Support Column
146	Diffuser
162-F	First Stage Impeller Retaining Collar
162-S	Second Stage and Above Impeller Retaining Collar
71	Nozzle Head
175	Suction Ball
190-L	Lower Sleeve Bearing
190-U	Upper Sleeve Bearing
218	Tank
223	Spacer Column
237-F	First Stage Snap Ring and Above
237-S	Second Stage Snap Ring and Above
257-L	Lower Gasket
257-I	Intermediate Gasket
257-U	Upper Gasket
257-S	Blind Flange Gasket
7	Shaft with Keys
910	Blind Flange

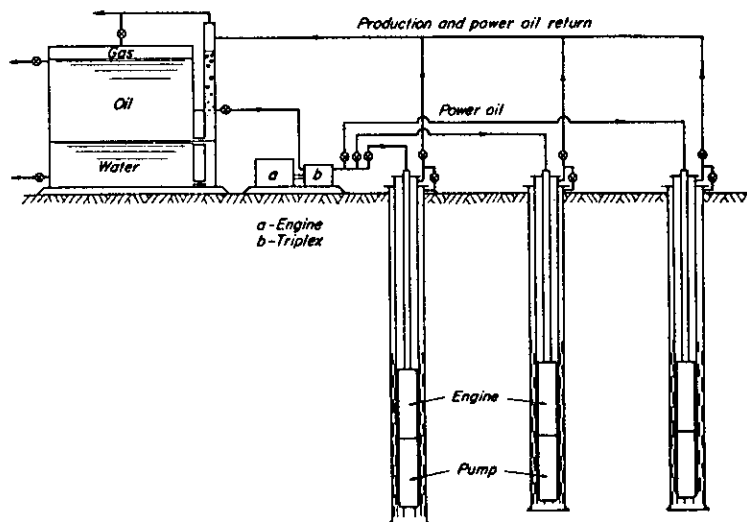
Figure 14.12 Four Stage Vertical Centrifugal Pump



(a) Rod Pump



(b) Centrifugal Pump



(c) Hydraulic Pump

Figure 14.13 Three Basic Types of Wellbore Pumps

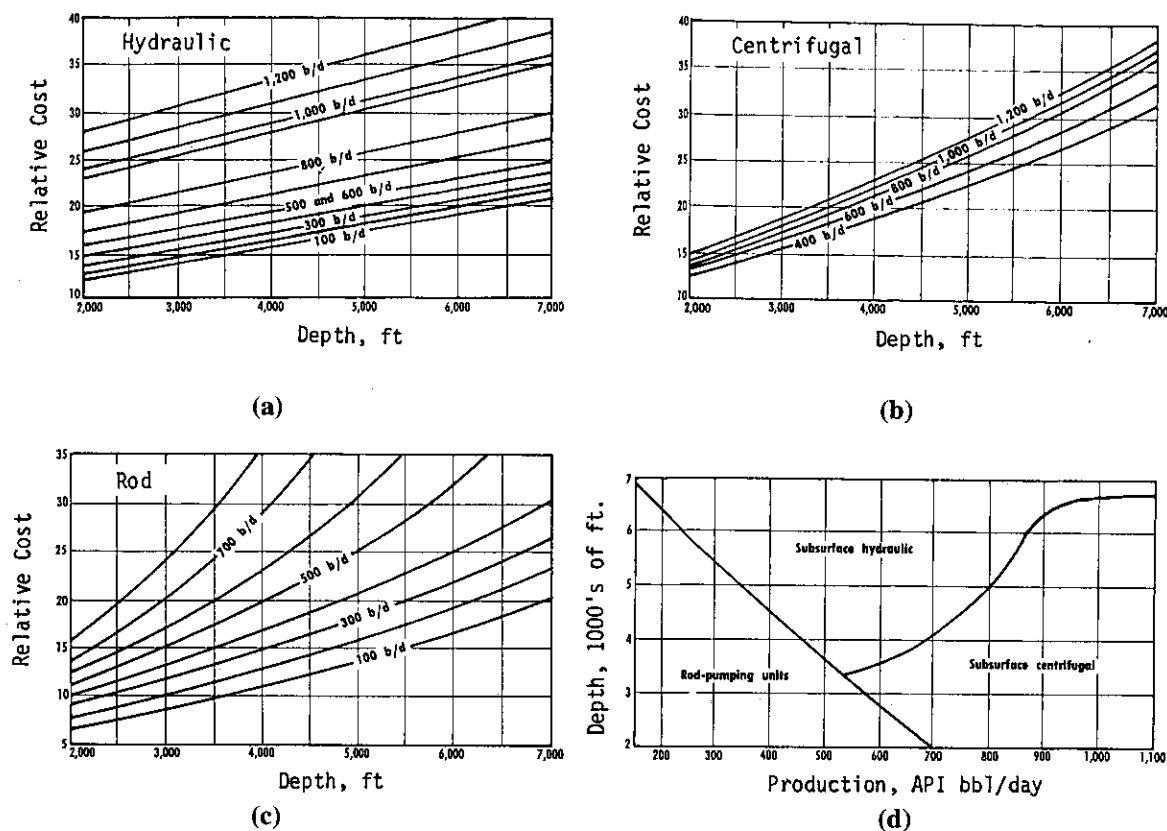


Figure 14.14 Relative Cost and Application of Wellbore Pumps

rate.^(14.14) Part (d) is a composite of Parts (a)-(c) and shows the optimum investment cost range for each type. The specific results shown apply only to one study, but the general pattern shown is typical.

The relative investment cost shown is for comparison purposes. A given value on the ordinate represents a comparable cost for the units. Absolute cost will vary with many factors.

HYDRAULIC EXPANDERS

In some instances a lot of power can be recovered by expanding liquid across an expander instead of a valve. This expander then serves as an "engine" to produce power or to drive other equipment.

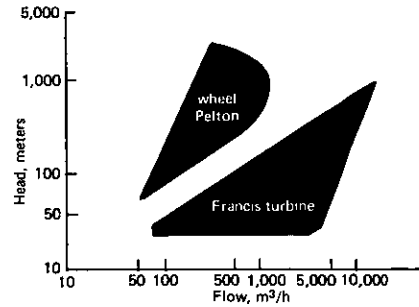
All rotating pumps are "free wheeling" and can run backward. Centrifugal and gear pumps have proven very satisfactory as expanders. In addition, there are several types of hydraulic turbines manufactured specifically for that service.

In a centrifugal pump the liquid enters at what normally would be the outlet and leaves at the "eye." A backward running pump has several critical limitations. Its efficiency may be about 10% less than a special power recovery turbine, even though its first cost is less. Also, control is more complicated, for you must contend with some independent control elements. With a flashing liquid, loss is emphasized in a reverse running pump because the impeller tends to shed the gas which bypasses the liquid.

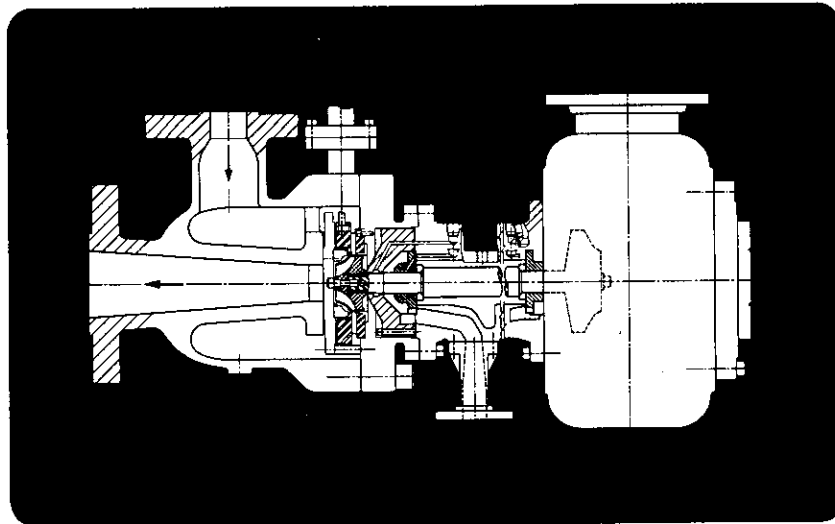
The performance curves of a centrifugal pump are different when used as a turbine. Efficiency is several percent lower and occurs at a higher capacity. Heat absorbed is greater than head developed in pump duty as is power development. Manufacturers have developed correction factors to predict pump performance as a turbine.

Three-screw gear expanders are very flexible, possess little vibration and noise, and deliver a 70-85% overall efficiency. Differential pressures to 2500 psi can be accommodated while delivering up to 450 kW.

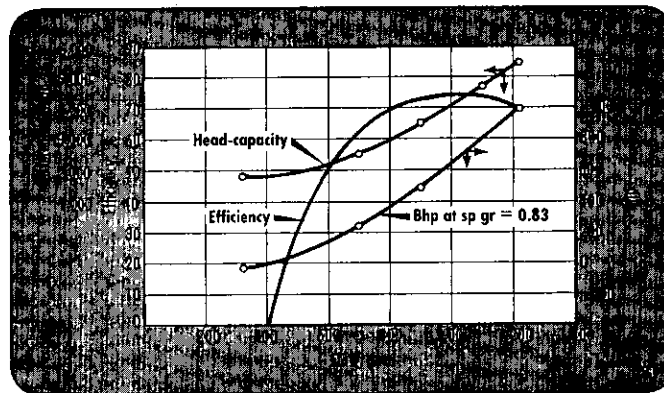
There are two types of regular hydraulic turbines – a Pelton wheel and a Francis runner. The Pelton wheel has a series of jets which impinge directly on a rotating impulse wheel. As shown at right, it is used primarily for high head, low flow applications.



The Francis runner uses wheels shown earlier for centrifugal pumps. Shown below is a cutaway of the Francis type expander.



The figure below shows performance of one turbine. The curve shapes are typical, although absolute values must vary with the specific model.



If the expander is run by liquid from a vessel where a level is being maintained, a level control is used.

References 14.15-14.18 review hydraulic expanders in more detail.

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15

COMPRESSORS

Compression of gases involves equipment comparable to that used for pumping liquids. Centrifugal and reciprocating compressors do the bulk of natural gas compression and thus will be emphasized in this chapter.

Screw, lobe and vane type compressors are used primarily in specialized services. Lobe and vane compressors may prove useful, particularly when discharge pressures do not exceed about 200 kPa [30 psig] and the flow rate is not too large. The screw compressor can be utilized for higher discharge pressures. All three possess favorable weight and vibration characteristics. Primary uses include instrument air compression, low pressure gas gathering, refrigeration and closed cycle solid bed dehydration units.

Table 15.1 is a rating of various compressors and Table 15.2 various drivers. Notice that four different listings are used for reciprocating compressors. Two are for separable units using high speed (H.S.) and low speed (L.S.) engines. A separable compressor is one where the gas compressor is a separate unit directly coupled to, or belt driven by, a separate driver (often on a common skid). The integral compressor has an internal combustion engine driver on the same frame, integral with the gas compression cylinders. The differentiation between large and small is somewhat arbitrary but "small" may be considered an integral with a rating to about 1000 kW, about the maximum size it is feasible to skid mount and assemble in a factory. Above this size the compressor has to be grouted on concrete foundations and piping field assembled.

TABLE 15.1
Comparative Guidelines to Selection of Compressors^(15.1)

Compressor Type	Reliability	Initial Cost	Installation Cost	Efficiency	Maintenance Cost	Weight/Space	Length Run	Ease Movement	Remove Adaptability	Adaptability to Change of Conditions
Low Pressure Screw or Lobe	E	E	E	G	E	E	E	E	E	G
Low Pressure Sliding Vane	G-P	E	E	G	F	E	F	E	E	G
H.S. Recip. Separable	G	E	E	G	G	E	G	E	G	E
L.S. Recip. Separable	E	G-F	G-F	E	E	F	G	P	G	E
Integral Engine Comp. Large	E	G-P	G-P	E	E	F	E	G-P	G	E
Integral Engine Comp. Small	E	G-F	G-F	E	E	G	E	G-F	E	G
Centrifugal	E	E-F	E	E-G	E	E	E	G-P	E	F-P

TABLE 15.2
Comparative Guidelines to Selection of Compressors Gas Compressor Drivers^(15.1)

Driver Type	Reliability	Initial Cost	Installation Cost	Efficiency	Maintenance Cost	Weight/Space	Length Run	Ease Movement	Remove Adaptability	Adaptability to Reciprocating	Adaptability to Rotary Compressor
High Speed Reciprocal Engine	G	E	E	G	G	E	G	E	G	E	F
Integral Eng. Comp.	E	G-P	G-P	E	E	F	E	P	G	E	P
Gas Turbine	E	G-P	E-F	G*	E	E	E	E	E	F	E
Steam Turbine	G	G	G	G-P	E	E	E	P	F	F	E
Electric Motor	E	E-G	E	E	E	E	E	E-P	E	G	E

Rating Symbols

E - Excellent

G - Good

F - Fair

P - Poor

*Thermal Efficiency can be increased by use of Recuperator and Waste Exhaust Heat Recovery

Exhaust Heat Recovery can also be utilized sometimes on Reciprocating Engines.

Figure 15.1 is a general representation of the application range of most of the compressors shown. The inlet flow rate is in actual m^3/minute at suction P and T. There is a broad overlap shown. Oftentimes the choice is made on economics and the mechanical factors involved.

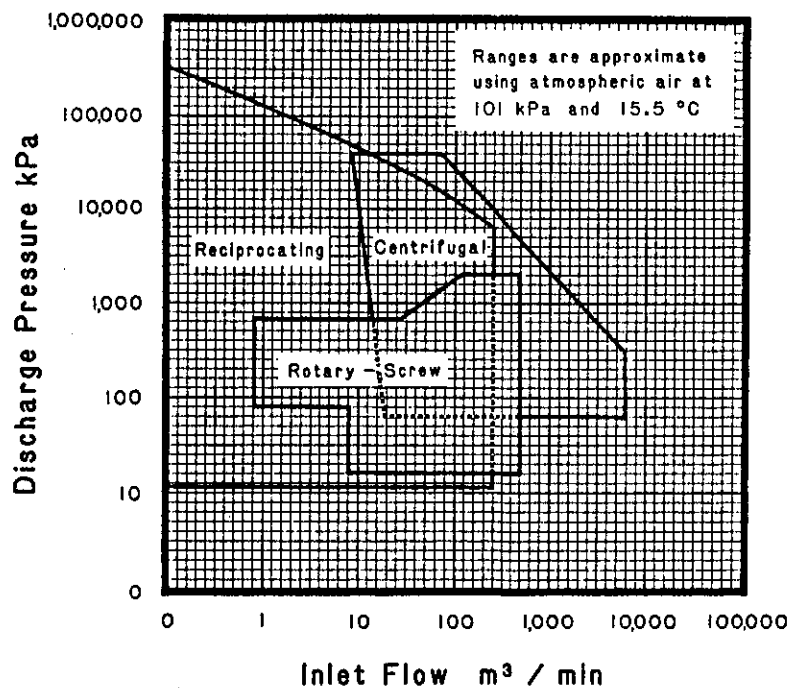
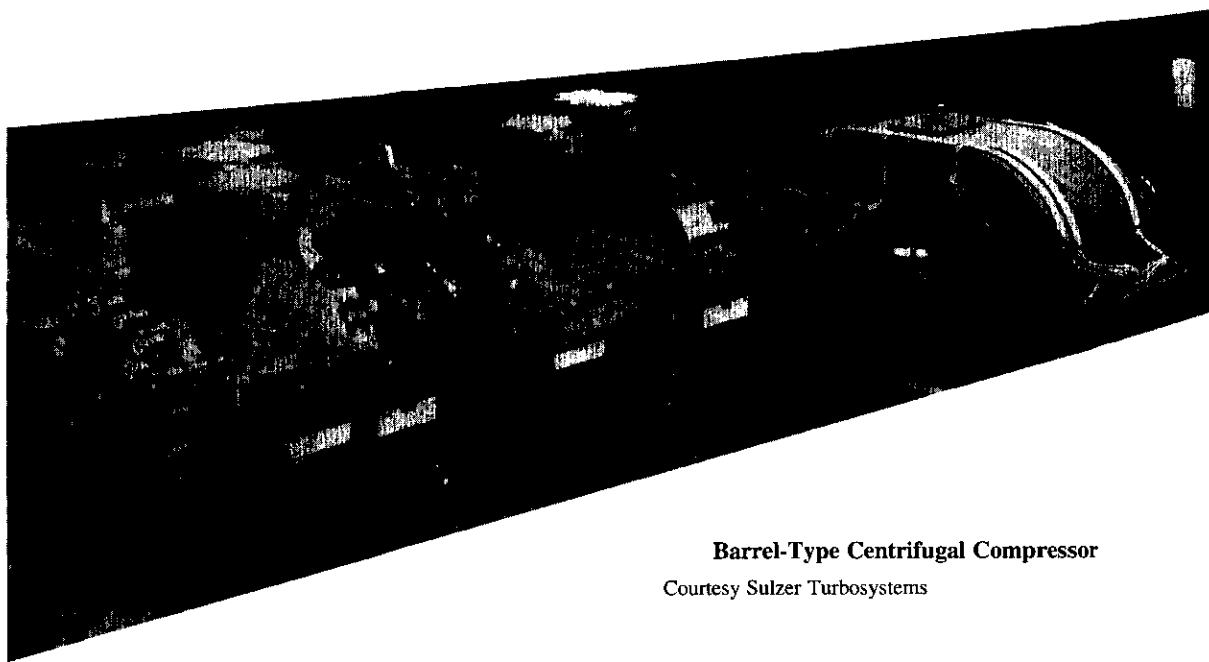


Figure 15.1 General Range of Application of Compressors

The centrifugal compressor is the most common choice. It produces the most power per unit weight and volume, and is less expensive per unit of power output. It is ideal for high flow rate and low to high head situations. In offshore and frontier area applications, the centrifugal is the usual choice. For all of these advantages, a centrifugal is not always the best choice. Its overall efficiency is lower than a reciprocal and



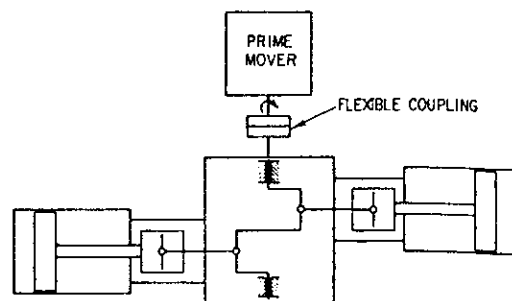
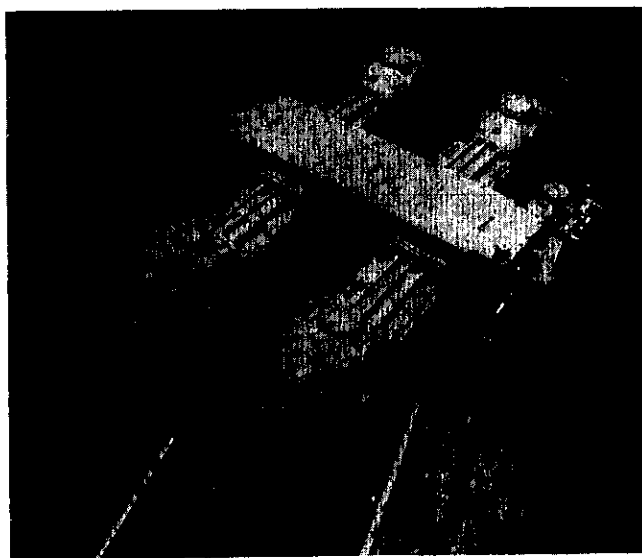
Barrel-Type Centrifugal Compressor

Courtesy Sulzer Turbosystems

fuel consumption is higher. It also is less "forgiving" of errors in specification and changes in operating conditions.

Unless the changes in gas rate and gas composition have been accurately forecast, the centrifugal unit may be forced to operate (if at all) outside its satisfactory range. Liquids, solids and other contaminants can prove troublesome. At high speeds, erosional tendencies are amplified. Centrifugal compressors are relatively simple and provide very satisfactory service when carefully chosen and applied with proper controls. The majority of the problems are people-oriented and not the fault of the machine.

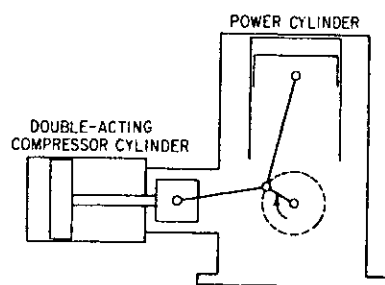
If you are considering a reciprocating unit, what type may be most desirable? In production operations when boosting small quantities of gas, a high-speed separable with an engine speed of 900+ rpm may be chosen. In small sizes, to several hundred kW, a small belt-driven unit may be suitable. For larger units a coupled, horizontally-opposed unit might be used. Such separables can be driven by turbines through speed reduction. Compressor units operating above 900 rpm are available to about 2300 kW. Units operating below 900 rpm are available to 12 000 kW. These slower speed separables may prove more attractive than integrals because of the flexibility afforded in using electric motors or drivers other than internal combustion engines.



← **Separable Reciprocating Compressor**

Courtesy Cooper Industries

Large integral compressors operating in a speed range of 300-600 rpm have the highest efficiency and lowest fuel consumption. They also are more costly to buy and are heavier. In spite of this, there are many applications (on land) where this is the best choice. Long life, low maintenance and ease of operation are paramount considerations.



Integral Type Reciprocating Compressor →
Courtesy Cooper Industries



COMPRESSOR POWER REQUIREMENTS

The *theoretical* amount of energy needed to compress a given amount of a gas between specified suction and discharge conditions is independent of the compressor unit. The *actual* amount of energy used depends on the efficiency of the compressor and its driver.

The basic thermodynamic equation is written

$$\Delta H = \int V dP = -W_{\text{theor}} \quad (15.1)$$

on the basis that the process is reversible ($S_1 = S_2$) and adiabatic ($Q = 0$). The subscript "theor" signifies these are the assumptions being used.

The error in these assumptions is corrected by an overall efficiency represented by the symbol "E."

$$W_{\text{act}} = \frac{W_{\text{theor}}}{E} \quad (15.2)$$

The value of "E" is used as a decimal in Equation 15.2 although it often is expressed as a percentage. It includes the effect of the thermodynamic (isentropic) efficiency and the mechanical efficiency. *Mechanical efficiency* refers only to frictional and other mechanical losses, including those in any valves.

Another efficiency, called *polytropic efficiency* is sometimes used in lieu of an isentropic efficiency. It will be defined and discussed in a later section.

The first step is the calculation of theoretical work (power). Using an efficiency from an acceptable correlation, one can then obtain actual power needs. It is usual to calculate power per stage and then multiply by the number of stages to find the total for that unit.

Use of Enthalpy Correlations to Obtain Power Requirements

Most computer solutions use the equation:

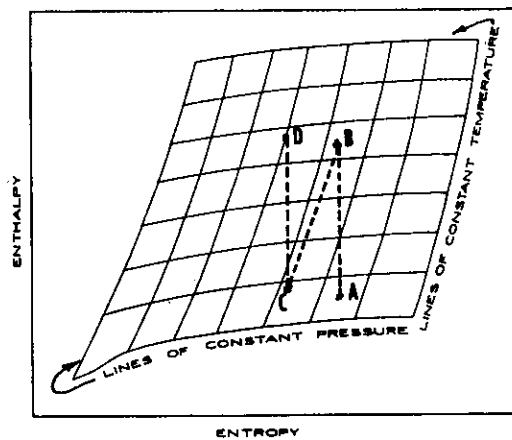
$$\text{Theor Power} = m (h_{2_{\text{isen}}} - h_1) \quad (15.3)$$

Where: m = gas flow rate
 h_2 = outlet enthalpy per unit mass
 h_1 = inlet enthalpy per unit mass

Needed is a subroutine to generate enthalpy values from an equation of state. Necessary data include inlet gas rate, pressure, temperature and composition plus outlet pressure. The corresponding outlet temperature is found from these data by assuming $S_2 = S_1$, using an iterative process. Δh_{isen} can now be calculated to find theoretical power needs.

The figure at left illustrates the procedure. Line AB is for an isentropic compression. With a reciprocating compressor it will be one stage; with a centrifugal it can be one stage or that number of stages in series before gas cooling is required. Line BC is a gas cooling step. Line CD represents a subsequent stage or stages, if any.

Figure 15.2 (a & b) can be used to find an approximate Δh_{isen} in a manual calculation for a typical 0.65-0.75 relative density gas. This figure also can be used to estimate the outlet temperature for a specified set of conditions.



It is included primarily to illustrate enthalpy calculation principle. For a manual calculation, one of the following methods generally is used.

The general equation is:

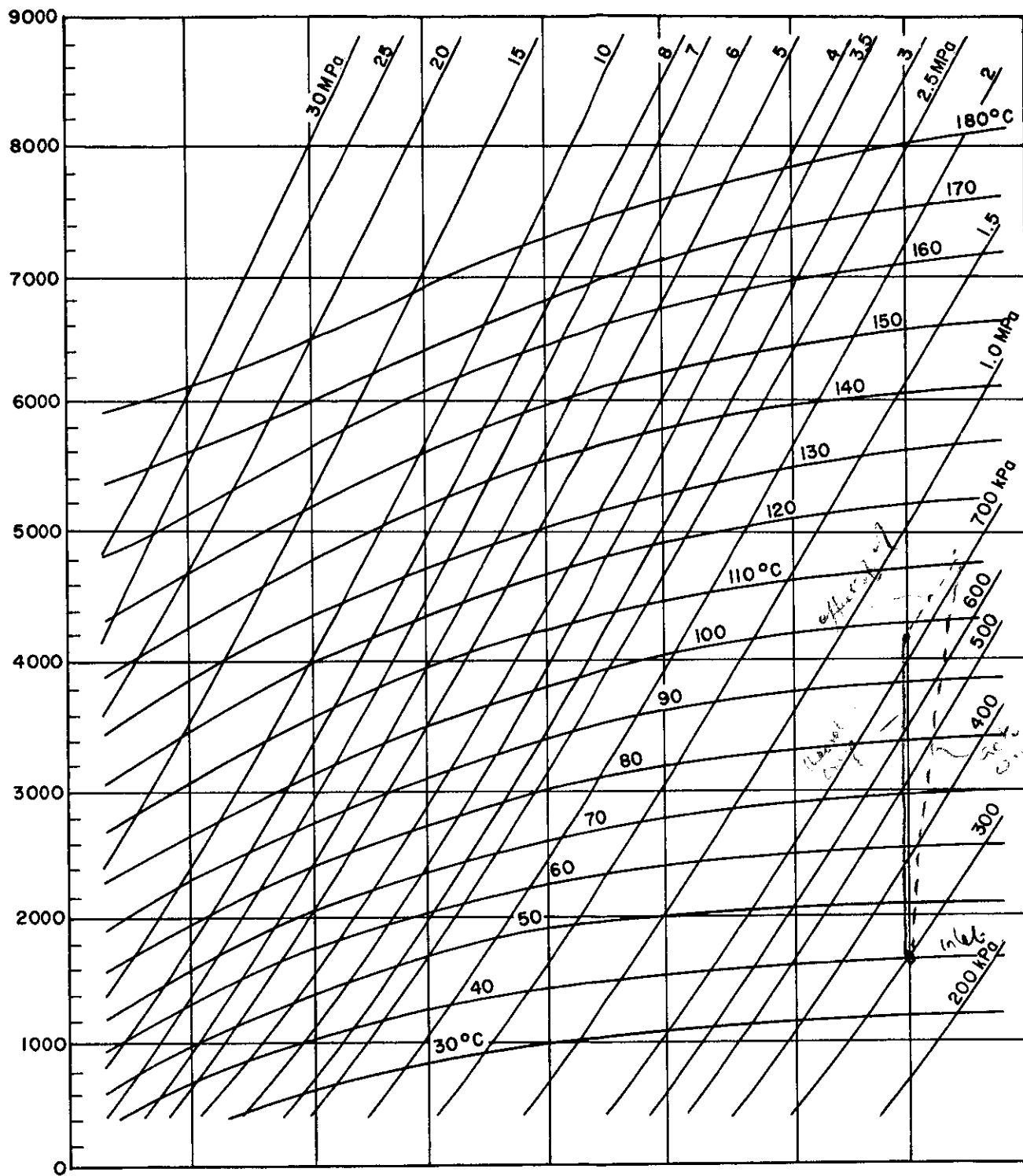
$$\text{Power} = \frac{(\text{Mass of Gas per Unit Time})(\Delta h \text{ per Unit Mass})}{(\text{Efficiency})(\text{Energy Conversion Factor})} \quad (15.4)$$

The common units employed are shown in the table below.

Power	Mass	Δh	Energy Conv. Factor
kW	(kg or kmol)/s	kJ/kg or kJ/kmol	1.0
kW	(kg or kmol)/h	kJ/kg or kJ/kmol	3600
kW	(lbm or lb-mol)/hr	Btu/lb or Btu/lb-mol	3413
BHP	(lbm or lb-mol)/hr	Btu/lb or Btu/lb-mol	2545
BHP	lbm/min	ft-lbf/lbm	33000

Compressor efficiencies vary with compressor type, size, and throughput. They can only be determined (after-the-fact) by compressor test although compressor manufacturers can usually provide good estimates. For planning purposes the following values may be used.

	Efficiency (E)
Centrifugal	0.65-0.75
H.S. Reciprocal	0.65-0.75
L.S. Reciprocal	0.75-0.85



① Actual $\Delta h = \frac{2800}{8} = 350$

Actual $p_2 = 1700 \times 3.25 = 5525$

\therefore plotting gives $p_2 = 112^{\circ}\text{C}$

Entropy kJ/kmol

① Say 2.5:1 compression

② $h_1 = 1700$ kJ/kmol
 $h_2 = 4200$ kJ/kmol
 $\Delta h = 2500$ kJ/kmol

Figure 15.2(a) Enthalpy-Entropy Diagram for a 0.65-0.75 Relative Density Sweet Natural Gas

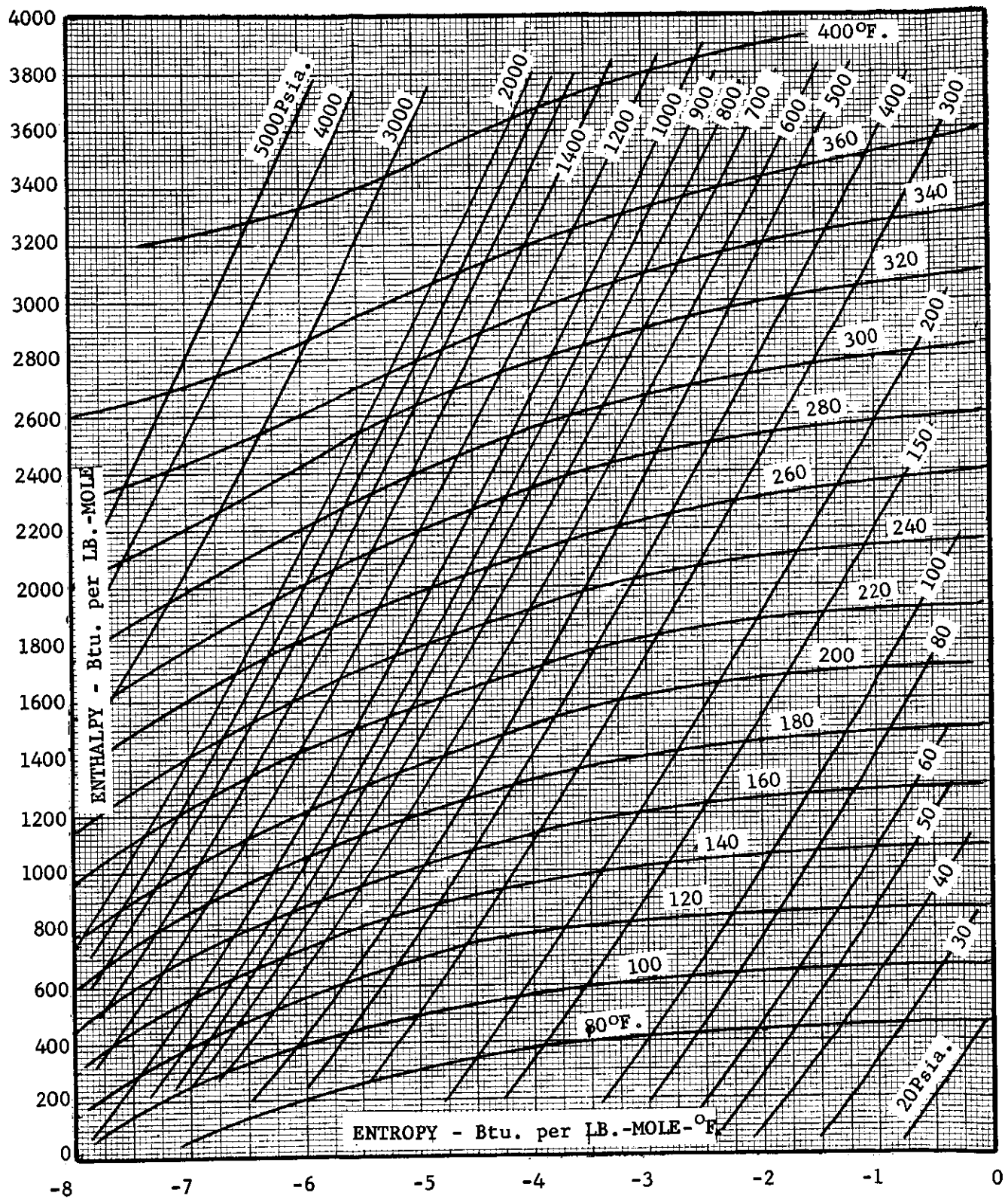


Figure 15.2(b) Enthalpy-Entropy Diagram for a 0.65-0.75 Relative Density Sweet Natural Gas

The above values represent overall efficiencies. Overall efficiencies include the mechanical losses (bearings, seals, gearbox, etc.). The mechanical efficiency varies with compressor size and type but 95% is a useful planning number.

$$\text{Overall Efficiency} = (\text{Isentropic Efficiency})(\text{Mechanical Efficiency})$$

Metric: $\text{kmol/h} = 1739 (10^6 \text{ std m}^3/\text{d})$

$$\text{kg/h} = 50\,372 (10^6 \text{ std m}^3/\text{d})(\gamma) = 1739 (10^6 \text{ std m}^3/\text{d})(M \text{ of gas})$$

English: $\text{lb-mol/hr} = 110 (\text{MMscf/d})$

$$\text{lb/hr} = 3178 (\text{MMscf/d})(\gamma) = 110 (\text{MMscf/d})(M \text{ of gas})$$

The previous equations are based on the following conversion factors:

Metric: Density of air = 1.21 kg/m^3 (standard conditions are 100 kPa and 15°C)
 $10^6 \text{ std m}^3 = 41\,740 \text{ kmol}$

English: Density of air = 0.0764 lb/ft^3 (standard conditions are 14.696 psia and 60°F)
 $1 \text{ MMscf} = 2636 \text{ lb-mol}$

In converting from Δh to power the following *conversion factors* are convenient:

$$1 \text{ hp} = 0.746 \text{ kW} = 2685 \text{ kJ/h} = 2545 \text{ Btu/hr}$$

$$1 \text{ kW} = 1.34 \text{ hp} = 3600 \text{ kJ/h} = 3413 \text{ Btu/hr}$$

Calculation of Power from the P-V Integral

For adiabatic compression of an *ideal gas* one can use the expression below to solve the term $\int V dP$.

$$PV^k = \text{constant}$$

where k is the ratio of specific heats, C_p/C_v . This ratio is convenient, for it is relatively constant with temperature for an ideal gas.

If one substitutes the above solution for $\int V dP$ into Equation 15.1, and inserts a compressibility term to correct the ideal gas assumption, an approximation for Δh_{isen} (from enthalpy data) results.

$$\Delta h_{\text{isen}} \approx \frac{T_1 Z_a R}{\left(\frac{k-1}{k}\right)(MW)} \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] \quad (15.5)$$

Where: Δh_{isen} = isentropic enthalpy change (head)

T_1 = suction temperature

Z_a = average compressibility factor $(Z_1 + Z_2)/2$

R = gas law constant

k = ratio of heat capacities (C_p/C_v)

MW = gas molecular weight

(P_2/P_1) = compression ratio

Metric	English
(see Table 15.3)	(see Table 15.3)
K	°R
—	—
(see Table 15.3)	(see Table 15.3)
—	—
kg/kmol	lb/lb-mol
—	—

TABLE 15.3
Values of Δh and R for Equation 15.5

Head	R
kJ/kg	8.314 kJ/kmol·K
m	848 kg·m/kmol·K
ft·lbf/lbm	1545 ft·lbf/lb-mol·°R
Btu/lbm	1.99 Btu/lb-mol·°R

The value of Z_a is sometimes found from the expression $(Z_1 + Z_2)/2Z_1$, as the best fit of actual data. However, the arithmetic mean gives consistent, realistic values. The values of Z may be found by one of the methods of Chapter 3 or from Figure 15.3. As noted, this figure is an approximation but is useful for natural gas manual compression calculations.

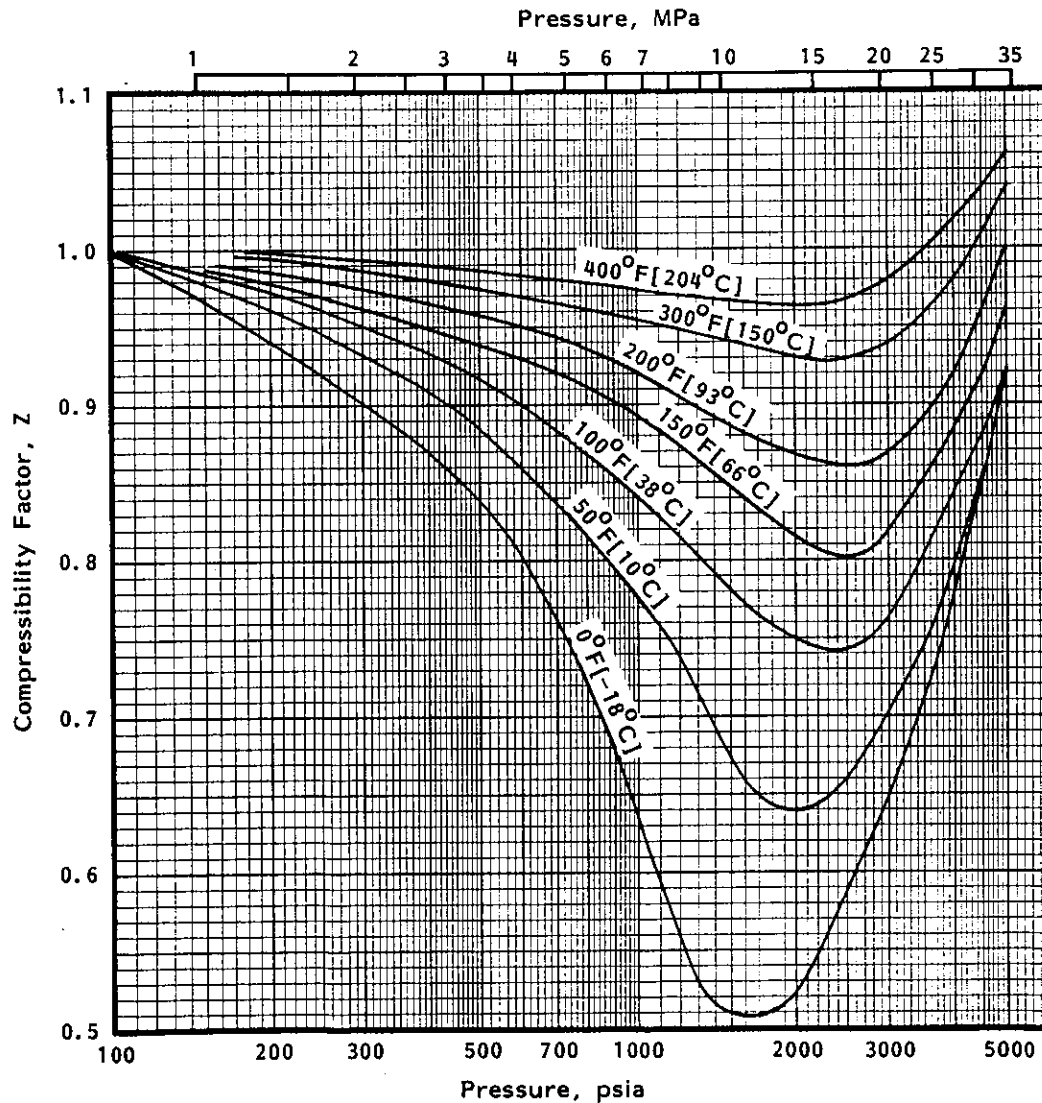


Figure 15.3 Approximate Z Chart for a 0.65-0.75 Relative Density Sweet Natural Gas for Use in Compressor Calculations

The value of "k" is found from Equation 15.6 when gas composition is known.

$$k = \frac{\sum (y_i)(C_{p_i})}{\sum (y_i)(C_{p_i}) - 1.99} \quad (15.6)$$

Where: y_i = mol. fr. of each component
 C_{p_i} = molar heat capacity from Table 15.4

(Note: If heat capacity is in kJ/(kmol·°C), substitute 8.314 for 1.99 in Equation 15.6.)

TABLE 15.4
Molar Heat Capacity of Hydrocarbons, Btu/(lbmol·°R)

Component	Temperature - °F						
	0	50	100	150	200	250	300
Methane	8.23	8.42	8.65	8.95	9.28	9.64	10.01
Ethane	11.44	12.17	12.95	13.78	14.63	15.49	16.34
Propane	15.65	16.88	18.17	19.52	20.89	22.25	23.56
i-Butane	20.40	22.15	23.95	25.77	27.59	29.39	31.11
n-Butane	20.80	22.38	24.08	25.81	27.55	29.23	30.90
i-Pentane	24.94	27.17	29.42	31.66	33.87	36.03	38.14
n-Pentane	25.64	27.61	29.71	31.86	33.99	36.08	38.13
Heptane	34.96	38.00	41.01	44.00	46.94	49.81	52.61
Octane	39.90	43.32	46.74	50.16	53.58	56.77	59.96
Nitrogen	6.95	6.95	6.96	6.96	6.97	6.98	7.00
Hydrogen Sulfide	8.00	8.09	8.18	8.27	8.36	8.46	8.55
Carbon Dioxide	8.38	8.70	9.00	9.29	9.56	9.81	10.05

The value of "k" may be approximated from the natural gas relative density.

$$k = 1.3 - (0.31)(\gamma - 0.55) \quad (15.7)$$

Equation 15.6 should be used when gas composition is known. Equation 15.7 is purely empirical, based on typical natural gases that contain no substantial quantities of contaminants and whose relative density does not exceed unity.

Combination of Equations 15.4 and 15.5 yields a convenient working equation for compression power

$$\text{kW/Stage} = \left(\frac{A}{E}\right) \left(\frac{k}{k-1}\right) (q) \left(\frac{P_s}{T_s}\right) (T_1) \left[\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} - 1 \right] (Z_a) \quad (15.8)$$

Where:

- A = conversion factor
- T_1 = suction temperature
- P_s = standard pressure
- T_s = standard temperature
- q = gas flow rate
- k = C_p/C_v
- $Z_a = (Z_1 + Z_2)/2$
- E = overall efficiency

Metric	English
11.57	2.26
K	°R
kPa	psia
K	°R
10^6 std m ³ /d	MMscf/d
—	—
—	—
—	—

When applied to a centrifugal compressor, Equation 15.8 may be applied for any number of stages between coolers. There is a practical limit to the gas temperature allowable in any compressor. Metallurgical, lubrication and efficiency factors are involved. If Equation 15.8 or an equivalent method is used, a separate determination of outlet temperature is required. Equation 15.9 below, can be used to estimate the discharge temperature.

$$T_D = T_1 \left[1 + \frac{[(P_2/P_1)^{k-1/k} - 1]}{E_{isen}} \right] \quad (15.9)$$

The efficiency term in the denominator of Equation 15.9 is the isentropic efficiency. For centrifugal compressors a value of 0.7 to 0.75 is suitable. For reciprocal compressors use 0.7 to 0.75 for high speed units and 0.83 to 0.90 for low speed.

The suction temperature to the next stage depends on the effectiveness of the interstage cooler, if one is used. For aerial cooling this temperature will likely be about 14-16°C [25-30°F] above the air dry bulb temperature used for design. For water cooling it will be about 12-14°C [20-25°F] above the air wet bulb.

Power from "Quickie" Charts

The above two basic calculation methods have been used to prepare some correlations to satisfy the need for a quick, easy approximation of power requirements. Two of these are summarized herein.

Figure 15.4 (a & b) is an example of a simple correlation that can be used for estimating compressor power. As a general rule, this figure will give slightly higher results than more exact methods.

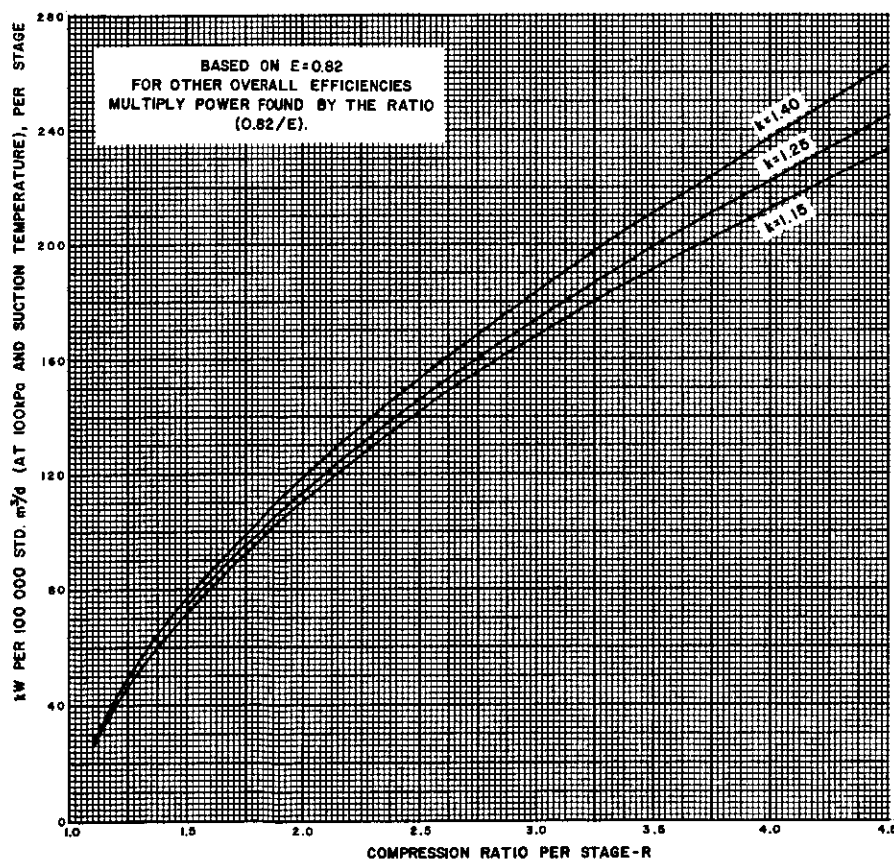


Figure 15.4(a) Simple Correlation for Estimating Compressor Power

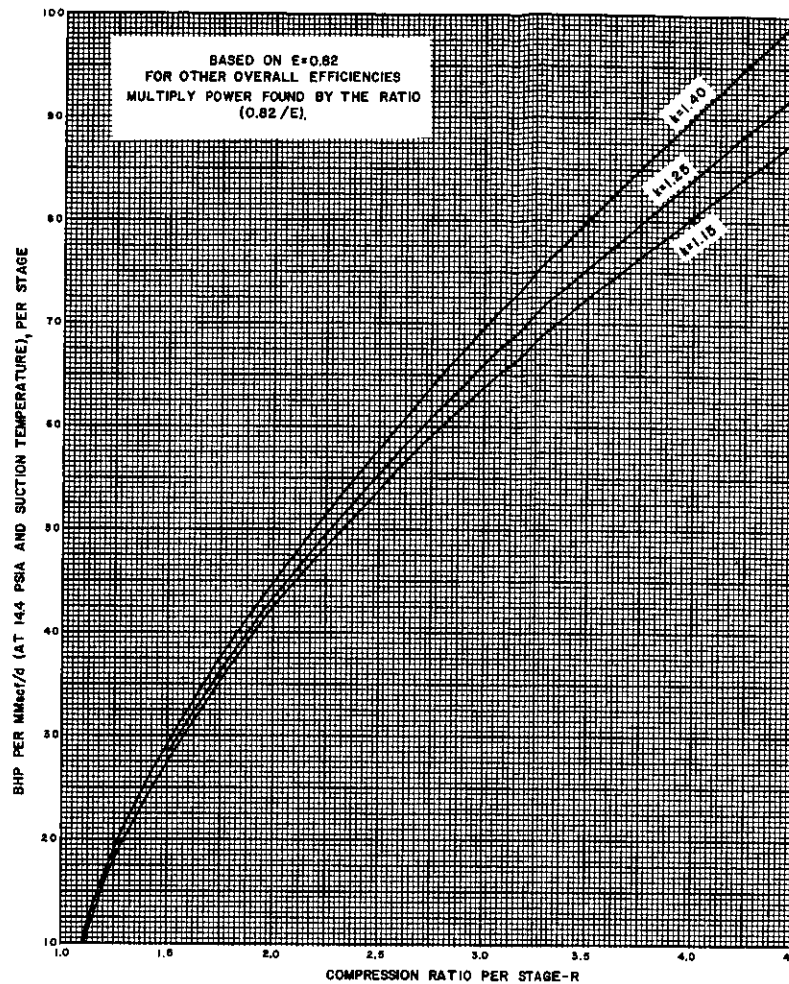


Figure 15.4(b) Simple Correlation for Estimating Compressor Power

The volumetric flow rate shown on the ordinate is at suction temperature, not at standard temperature. So ... the volumetric flow rate of gas at standard conditions must be converted to the conditions specified by the following equation.

Metric:
$$q \text{ (multiplier for chart)} = \frac{\text{std m}^3/\text{d}}{100\,000} \left(\frac{P_s}{100} \right) \left(\frac{T_1}{T_s} \right) \quad (15.10)$$

Where: $\text{std m}^3/\text{d}$ = flow rate at 100 kPa and 15°C
 T_s = K
 T_1 = suction temperature, K
 P_s = kPa

English:
$$q \text{ (multiplier for chart)} = \text{MMscf/d} \left(\frac{P_s}{14.4} \right) \left(\frac{T_1}{T_s} \right) \quad (15.11)$$

Where: MMscf/d = flow rate at P_s and T_s
 T_s and T_1 = °R, P_s = psia

The power from Figure 15.4 is multiplied by the adjusted "q" to find power per stage. The charts are for $E = 0.82$. They may be used for any compressor efficiency by adjusting for the efficiency of the unit involved.

Figure 15.5 is a nomograph for estimating BHP. It is for an overall efficiency of 0.73. For another efficiency multiply the BHP found by $(0.73/E)$.

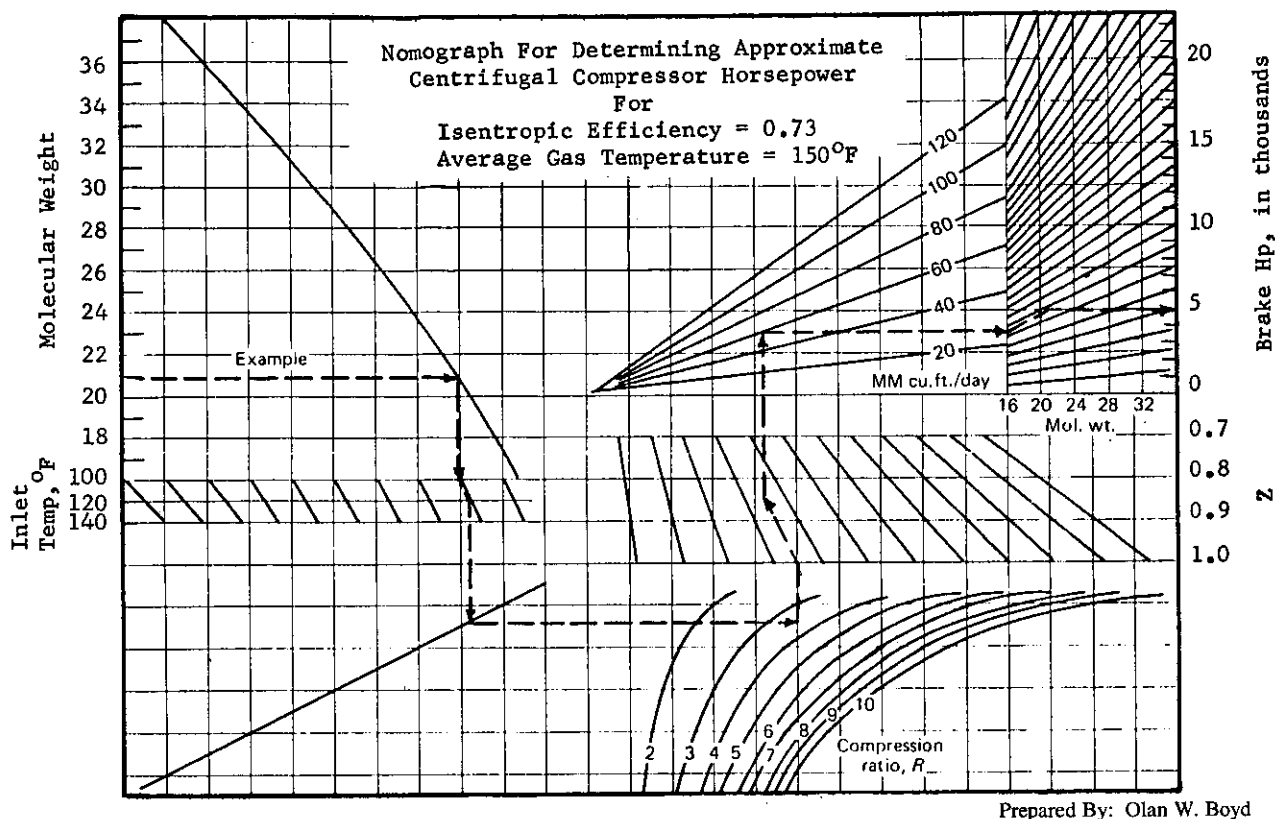


Figure 15.5 Nomograph for Estimating Compressor Power

Simple correlations like those shown in Figures 15.4 and 15.5 are as useful for early planning purposes as more complicated computer solutions. Flow rates and conditions are more uncertain than the calculation. In reality, the calculation of compressor power requirements is the most routine of all the considerations involved. The calculation and selection of the compression end details – and the control system – usually are most critical to obtain a satisfactory unit.

For planning purposes, before a specific compressor and driver has been chosen, $E = 0.82$ for low speed reciprocating units and $E = 0.73$ for centrifugal units are suitable, safe values. In dealing with a vendor, be certain which efficiency is being quoted. It is not so important which efficiency is being used so long as the communication is clear.

However you determine it, the power requirement is used to pick a standard, discrete size of unit. Normally, it will be a unit capable of producing more power than that calculated under the most adverse compression conditions anticipated. Because of size or possible flow fluctuations, units in parallel may be considered. The power calculation provides a guideline for considerations like these.

Isentropic Versus Polytropic Efficiency

Traditionally compressor vendors have used polytropic rather than the isentropic efficiency when quoting compressor performance. This is because polytropic efficiency $[E_{poly}]$ is essentially independent of compression ratio and gas composition. Polytropic efficiency is based on an imaginary "polytropic path" which is reversible and non-adiabatic (isentropic is reversible and adiabatic). For a polytropic path, the exponent "n" is substituted for "k" in Equations 15.5 and 15.8. In Equation 15.8 the "overall" efficiency would then be

based on E_{poly} rather than E_{isen} . Polytropic efficiency can only be determined by compressor test. Some useful approximate relationships between polytropic and isentropic equations are shown below.

$$\frac{n-1}{n} \approx \frac{k-1}{k E_{poly}} \quad (15.12)$$

$$E_{isen} \approx \frac{\left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right]}{\left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]} \quad (15.13)$$

Equation 15.13 is shown graphically in Figure 15.6.

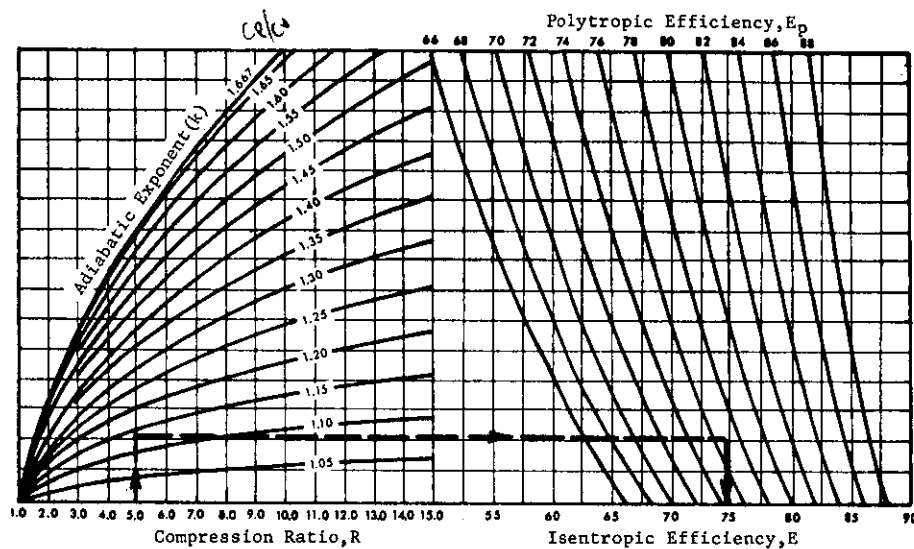


Figure 15.6 Relationship Between Thermodynamic and Polytopic Efficiency

Both approaches produce the same basic result. Process engineers tend to favor the isentropic method while compressor specialists favor polytropic.

CENTRIFUGAL AND AXIAL COMPRESSORS

Centrifugal compressors should be considered when actual inlet flows exceed about $0.1 \text{ m}^3/\text{s}$ [200 ACFM]. They offer the advantages of more power per unit weight and they operate essentially vibration free. This makes them particularly attractive for offshore locations or where air transportation to remote locations is necessary. Their initial cost normally is less than for reciprocating compressors but the efficiency is less and utility costs may be higher. In addition, the ability of a centrifugal compressor to handle changing condition (pressures, temperature, molecular weights) may be limited.

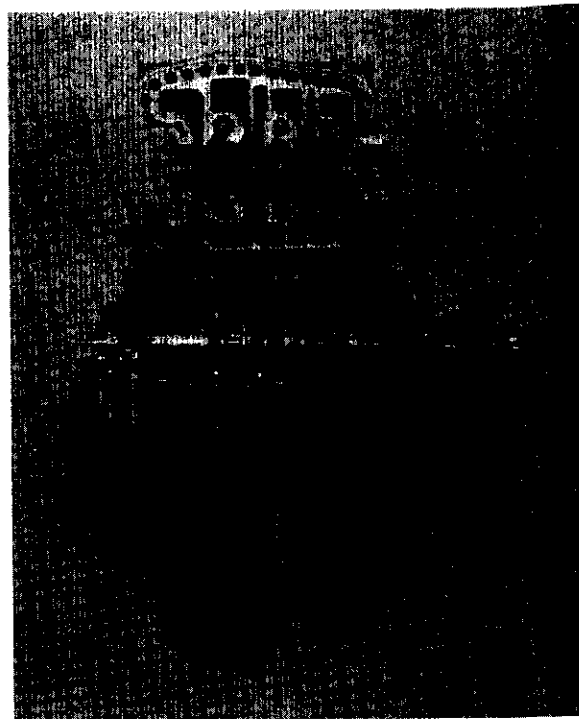
Centrifugal compressors are manufactured in three configurations:

1. overhung impeller
2. horizontally split
3. vertically split (barrel-type)

Overhung impellers are typically used in single stage service. The booster compressor in a turboexpander unit is a good example of this type of compressor. The impeller is usually open, backward-bladed.

Horizontally split cases are often used in high volume lower pressure applications. Figure 15.7 shows a horizontally split case. These compressors have casings split horizontally at the mid-section. The halves are bolted and doweled together. The internal parts such as the rotor, bearings and seals are accessible for repair and inspection by removing the top half. To facilitate access compressor nozzles are often oriented downwards from the bottom half of the case.

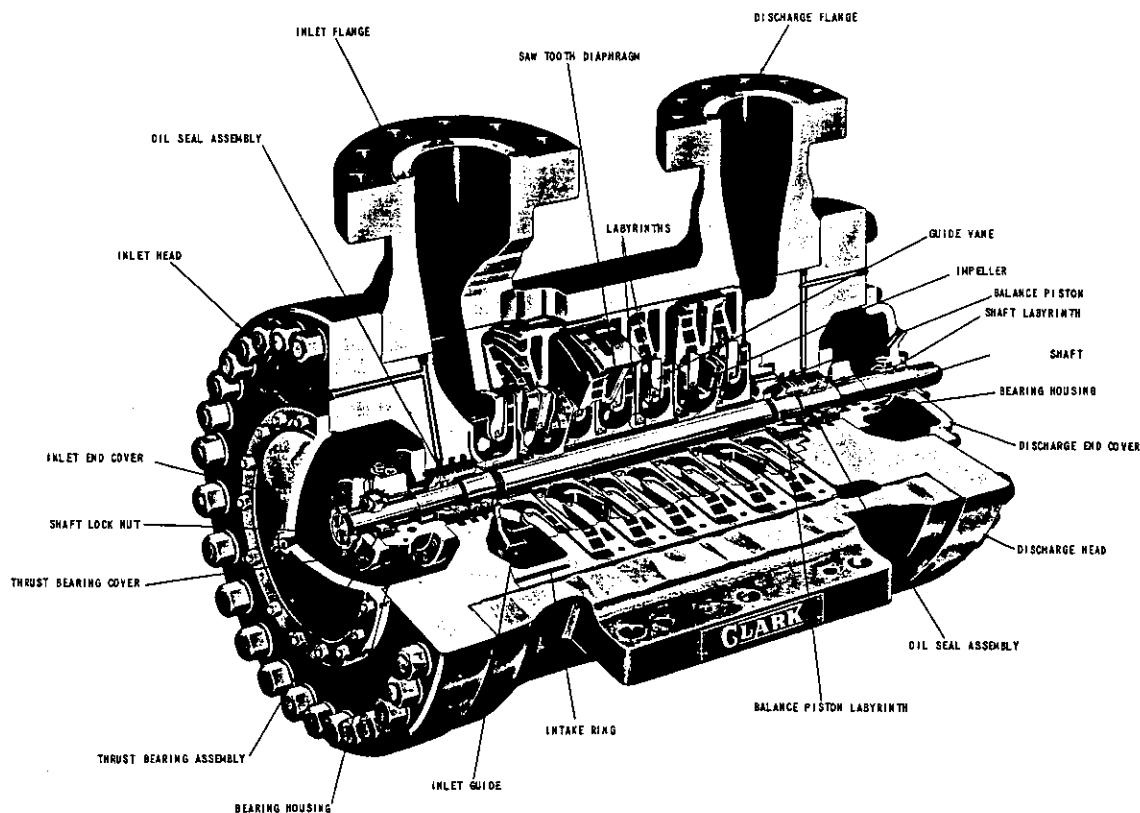
In vertically split or barrel type compressors the compressor "barrel" contains the rotor and diaphragms and is removed from the end of the compressor. These compressors are typically used in higher pressure/low volume applications. If a spare barrel is available they can be repaired more quickly than horizontally split compressors. They require additional plot area since the space must be provided to remove the barrel from the case (similar to a shell and tube heat exchanger).



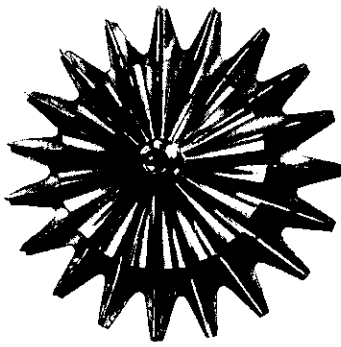
Courtesy of Sulzer Turbosystems

Figure 15.7 Horizontally Split Case Centrifugal Compressor

Shown below is a cut-away portion of a vertically-split centrifugal compressor using a closed, backward-bladed impeller, the most common type in natural gas operations.



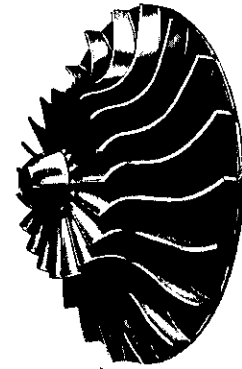
The form of impeller blade affects the characteristic performance curve, the operating range and the efficiency of a centrifugal compressor. Shown below are three types.



Open Radial-Bladed Impeller



Closed Backward-Bladed Impeller



Open Backward-Bladed Impeller

The same basic specific speed and diameter relationships used for pumps in Chapter 15 apply also for compressors, particularly centrifugal and axial. Figure 15.8 is based on the equations:

$$d_s = \frac{(A)(d)(H)^{0.25}}{q^{0.5}}, \quad N_s = \frac{(B)(N)(q)^{0.5}}{H^{0.75}} \quad (15.14)$$

Where:

- d_s = specific diameters
- N_s = specific speed
- d = diameter
- q = gas flow rate
- H = head per stage
- N = rpm
- A =
- B =

Metric	English
m	ft
m ³ /s	ft ³ /sec
m	ft
m ³ /s	ft ³ /sec
m	ft
—	—
0.74	1.00
2.44	1.00

Figure 15.8 shows the relationship between these quantities for various types of compressors. Normally one would not use a centrifugal when the efficiency is less than about 0.7 unless weight or vibration limitations dictate it. The number of stages and impeller diameter combinations must be chosen to achieve a satisfactory efficiency.

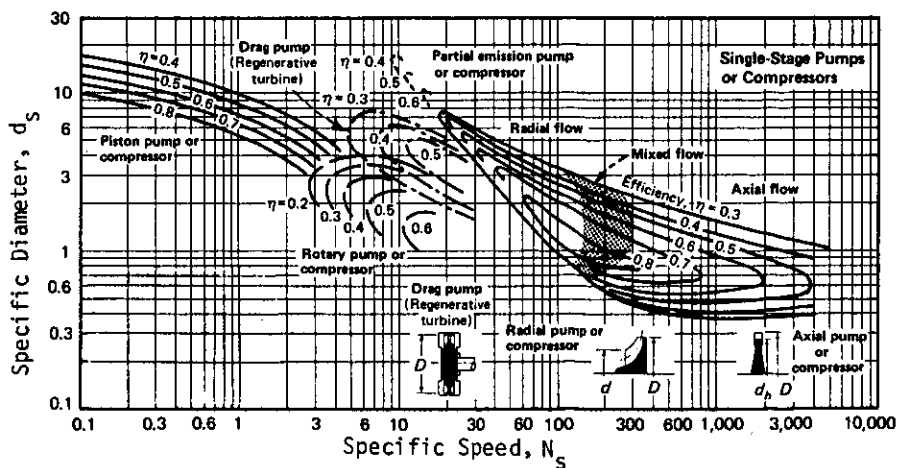


Figure 15.8 Relationship Between Specific Speed and Diameter for Compressors^(15.3)

Centrifugal Power Calculation

Each of the methods for power calculations shown previously can be used to estimate centrifugal compressor power. Typical polytropic efficiencies vary from 72 to 81% with 78% a good value for planning purposes in natural gas compression. Mechanical losses are normally estimated as follows:

Labyrinth seals and balance piston - The balance piston and labyrinth seals allow high pressure gas to leak back to suction pressure. The additional work required to recompress this internally bypassed gas is usually handled by increasing the design gas flow through the compressor when performing the power calculation. This increased flow usually varies from 2 to 5% of total flow with 3% a useful number for planning calculations.

Bearing and Seals - Mechanical losses at the bearings and seals are a function of compressor size and speed. Equation 15.15 provides a quick approximation of these losses.

$$W_L = F_L \left(\frac{N}{1000} \right)^2 \quad (15.15)$$

Where: W_L = mechanical losses, kW
 F_L = loss factor (see table at right)
 N = compressor speed, rpm

$q, \text{m}^3/\text{s}$	F_L
1.5	0.20
2.1	0.37
3.1	0.69
4.25	1.28
6	2.38
8.7	4.45
12.5	8.3
17	15.4

The foregoing guidelines are based in standard impeller design and oil seals. For dry seals, seal and bearing losses may be substantially smaller than indicated by the above.

Gearbox. - Typical efficiencies for speed reducing gears vary from 0.95 to 0.98. For planning purposes, 0.97 is a useful design number.

Number of Impellers and Speed

The energy imparted to the gas in a centrifugal compressor is termed the "head." The theoretical head may be calculated using an isentropic or polytropic path. The head is the specific enthalpy change of the gas across the compressor. As with centrifugal pumps it has been traditional to represent the compressor head in terms of meters or feet. The compressor head is related to the pressure rise by Equation 15.5.

The energy imparted to the gas in a dynamic compressor is related to velocity change through the impeller/diffuser. Like centrifugal pumps this energy change is a function of impeller tip speed and actual flow rate and is essentially independent of the gas compressed. The maximum impeller tip speed is determined by mechanical constraints but a value of 250-300 m/s [820-984 ft/sec] is typical for centrifugal compressors in sweet service. Equation 15.16 relates head per impeller to impeller tip speed.

$$\Delta h = \frac{\mu u^2}{2g} \quad (15.16)$$

Where: Δh = polytropic head/impellar
 μ = head coefficient
 u = impeller tip speed
 g = gravitational force

Metric	English
m	ft
0.8 to 1.10	
m/s	ft/sec
9.81 m/s ²	32.18 ft/sec ²

For multistage natural gas compression, typical design head coefficient is 1.0. Using a fairly conservative tip speed of 250 m/s [820 ft/sec] the maximum head which can be imparted to the gas across one impeller is approximately 3200 m [10 500 ft]. These values are useful for planning purposes to estimate the number of impellers required to achieve a given pressure rise across a compressor case.

For heavy molecular weight gases (such as refrigerants) the impeller tip speed may be limited by the sonic velocity. Sonic velocity in gases can be estimated from Equation 15.17.

$$u_s = \sqrt{\frac{g_c k Z R T}{MW}} \quad (15.17)$$

Where: u_s = sonic velocity
 k = heat capacity ratio (C_p/C_v)
 R = gas law constant
 T = gas temperature
 g_c = mass/force constant
 MW = gas molecular weight

Metric	English
m/s	ft/sec
-	-
8314	1545
K	°R
1.0	32.18

Impeller tip speed should not exceed 110% of u_s where u_s is determined at suction conditions.

Equation 15.18 can be used to estimate impeller diameter. It is based on backward-bladed impellers used in multistage compressors. It is consistent with the results of Equation 15.4 and Figure 15.8.

$$d = \sqrt{\frac{q}{0.050 u}} \quad (15.18)$$

Where: d = impeller diameter
 q = inlet flow rate
 u = impeller tip speed

Metric	English
m	ft
m ³ /s	ft ³ /sec
m/s	ft/sec

Compressor speed can then be determined

$$N = \frac{60 u}{d \pi} \quad (15.19)$$

Where: N = compressor speed
 u = tip speed
 d = impeller diameter

Metric	English
rpm	rpm
m/s	ft/sec
m	ft

For a centrifugal compressor

1. Head varies with rpm squared.
2. Flow varies directly with rpm (depending on compression ratio).
3. Power varies directly with rpm cubed.

Temperature must rise in accordance with equations shown in an earlier section. Inter-stage cooling may be necessary, but when used, any condensed liquids must be removed in a properly designed separator. Where feasible, some installations use spray injection between stages so that water or some other fluid may be injected directly into the gas stream. The potential problems with this should be obvious.

Water may be circulated through the diaphragms between stages. Cooling is limited by area and metal wall thickness. The most positive approach is external cooling. The gas is withdrawn from the case, externally cooled, and then re-enters.

Example 15.1: A centrifugal compressor is required to compress 2.5×10^6 std m^3/d [88 MMscf/d] of a 0.65 sp. gr. natural gas from 1500 to 4500 kPa [218-653 psia]. Suction temperature is 35°C [95°F] and the average compressibility factor is 0.95. The compressor efficiency (polytropic) is 78%. Calculate the following:

1. Compressor head
2. Compressor power
3. Compressor discharge temperature
4. Number of impellers required
5. Approximate impeller diameter
6. Approximate shaft speed

Use isentropic approach –

1. Compressor head (Equation 15.5)

$$\text{From Equation 15.7, } k = 1.3 - (0.31)(0.65 - 0.55) = 1.27$$

$$(k - 1)/k = 0.2126$$

$$\Delta h = \frac{(308)(0.95)(848)}{(0.2126)(0.65)(28.97)} \left[(3.0)^{0.2126} - 1 \right] = 16\,300 \text{ m}$$

2. Compressor power (Equation 15.8), $E_{\text{isen}} = 0.75$

$$\text{Actual gas rate} = (1.03)(2.5) = 2.575 \times 10^6 \text{ std m}^3$$

$$\text{kW} = \left(\frac{11.57}{0.75} \right) \left(\frac{1.27}{0.27} \right) (2.575) \left(\frac{100}{288} \right) (308) \left[(3.0)^{(0.27/1.27)} - 1 \right] (0.95)$$

$$= 4994 \text{ kW}$$

Bearing and Seal Losses

$$W_L = 0.37 (12\,000/1000)^2 = 53 \text{ kW}$$

$$\text{Total Power} = 4994 + 53 = \underline{5047 \text{ kW}}$$

3. Compressor discharge temperature

$$T_2 = 308 \left[1 + \frac{[(3.0)^{(0.27/1.27)} - 1]}{0.75} \right] = 416 \text{ K} = \underline{143^\circ\text{C}}$$

4. Number of Impellers (assume 3200 m/impeller)

$$\text{No. of impellers} = 16\,300/3200 \approx 5.0$$

5. Approximate impeller diameter (1st stage impeller)

$$q = \left(\frac{2.5 \times 10^6}{86\,400} \right) \left(\frac{100}{1500} \right) \left(\frac{308}{288} \right) (0.95) = 1.96 \text{ m}^3/\text{s}$$

From Equation 15.18

$$d = \left[\frac{1.96}{(0.050)(250)} \right]^{0.5} = \underline{0.4 \text{ m}}$$

6. Approximate shaft speed, N

$$u = 250 \text{ m/s}$$

$$N = (250)(60)/0.40 \pi = \underline{12\,000 \text{ rpm}}$$

Check E on Figure 15.8

$$D_s = \frac{(0.74)(0.4)(3200)^{0.25}}{(1.96)^{0.5}} = 1.6$$

$$N_s = \frac{(2.44)(12\,000)(1.96)^{0.5}}{(3200)^{0.75}} = 95$$

Surge and Choke

Surge occurs at some minimum capacity, at each speed, where unstable operation results. At this capacity, the compressor does not meet the discharge pressure so that a series of flow reversals occur. Surge occurs because the compressor alternately delivers gas and the system returns it. A given stage surges at a capacity somewhat proportional to gas molecular weight.

Choke (the "stonewall" effect) limits compressor capacity. This condition is caused by the limiting flow rate of the gas through the "eye" of the first impeller. This flow is always higher than design and usually would not occur below 115-120% of rated capacity. The maximum velocity is limited by the Mach number (speed of sound) of the gas. Theoretically, the choke effect would occur at this value, but usual practice is to limit design to 0.85-0.90 of the Mach number. The lightest gas compressed where choke has been a significant problem is propylene. Propane, butane, and the Freons have Mach velocities of about 200 m/s at minus 40°C. When compressing gases lighter than propane, "choke" is not of practical concern.

Characteristic Curves

Figure 15.9 shows the characteristic curve for a multistage centrifugal compressor. Both surge and choke lines are shown. The compressor must be operated between these stability limits.

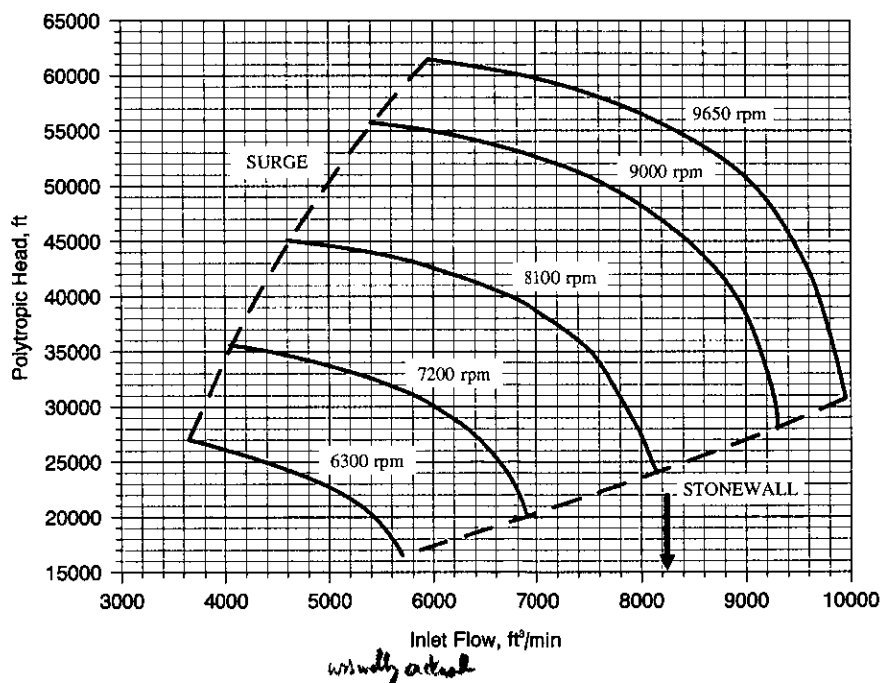


Figure 15.9 Typical Characteristic Curve for a Centrifugal Compressor

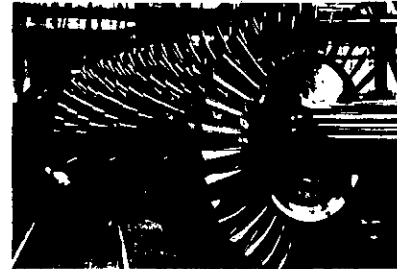
Notice that at a given rpm the head may not vary much with gas flow rate. This is why a centrifugal is sometimes called a constant head device. This affects control for varying flow rates. As flow rate decreases, there is no problem until it approaches the surge line. Then, something has to be done. One can reduce speed or bypass gas to keep the throughput artificially above the surge line. One cannot design the control system until one has the characteristic curve for the specific unit involved.

One needs to ascertain the likely magnitude and frequency of flow or head changes. It is possible that several units may be necessary in parallel. Although it may be mechanically feasible to operate at low power loads, poor economics results. In some cases, it is more economical to use multiple units and idle one or more.

AXIAL COMPRESSORS

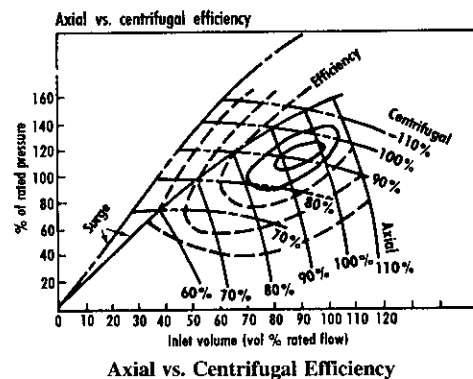
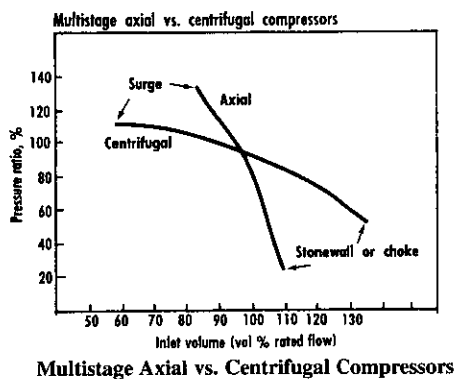
An axial compressor is similar in principle to a centrifugal. Rather than using wheels, a series of blades is involved – rotating (rotors) and stationary (stators). About half of the pressure rise is accomplished by the rotors and the other half by the stators.

The rotors turn and add kinetic energy to the gas. Each row of stators converts kinetic energy to pressure and acts as a diffuser for the gas flowing off the preceding row of rotors. The stators also serve as nozzles to guide the gas into the next row of rotors. Each stage consists of one row of rotors and one row of stators. Often about two stages are needed in an axial compressor for each one stage in comparable centrifugal.



The axial compressor is used primarily when flow rate is above 30 m³/s [60 000 ACFM], at suction conditions, and the discharge pressure is 3500-4000 kPa [500-580 psia] or less. In the range 30-60 m³/s [60 000-120 000 ACFM] there is an overlap between the two types. Choice will depend on pressure and other factors.

An axial will tend to have a higher efficiency than a centrifugal, about 8-10% higher. It is more compatible with a gas turbine with regard to thrust loading. For large flow, medium pressure duties it offers an excellent alternative to the centrifugal compressor.

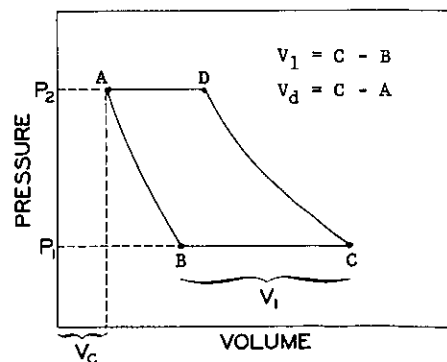


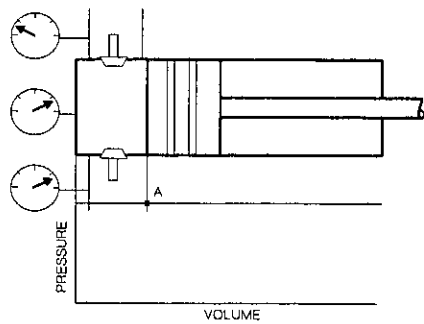
It may be controlled by speed, bypass, suction throttling or by use of variable angle guide vanes. As noted above, it is much less a constant head device than a centrifugal.

RECIPROCATING COMPRESSORS

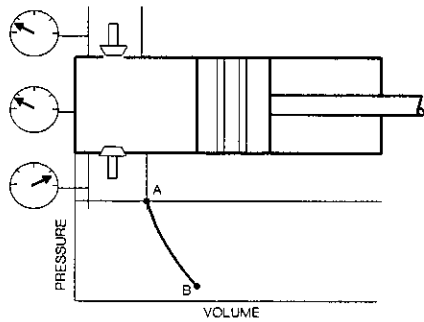
Figure 15.10 shows the basic performance of a reciprocating compressor cylinder. This is shown in greater detail in Figure 15.11.

Point A represents the end of the compression stroke. Line ABC represents the total suction stroke. Section AB of this line represents expansion of the gas trapped between the piston and the end of the cylinder at pressure P_2 . No new gas can enter the cylinder until this gas expands to pressure P_1 (point B). Volume V_1 (Line BC) represents the new gas entering on the suction stroke. The capacity of the cylinder is fixed by volume V_1 . This volume, in turn, depends on the compression ratio (P_2/P_1) and volume V_c .

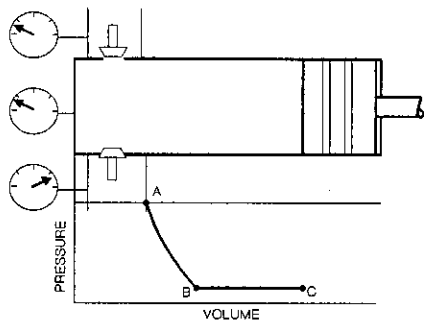




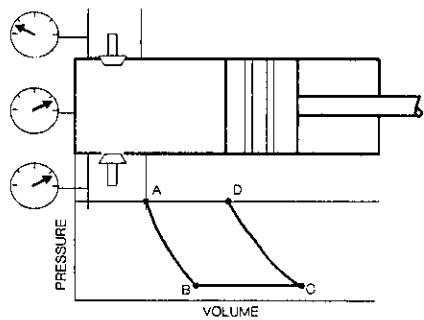
Start of cycle at point A. Piston has reached end of stroke and moves to the right. Suction and discharge valves closed. Cylinder filled with gas at *discharge* pressure.



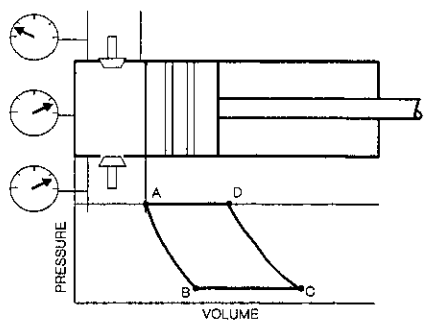
As piston moves to the right, the gas in the cylinder at the start begins to *expand* and its pressure drops. At point B, pressure in cylinder equals suction pressure and suction valve opens.



As piston moves from point B to C, suction valve remains open, and gas flows into cylinder. When piston starts to move to the left, the pressure in the cylinder rises and the suction valve closes.



When piston moves from C to D, pressure in cylinder rises. At point D, pressure reaches discharge pressure and discharge valve opens.



As piston moves from D to A, gas flows out of the cylinder into the discharge line. At point A, cycle starts over.

Figure 15.11 Reciprocating Compressor Compression Cycle

The volume represented by line ABC is known as the *piston displacement* – the volume of gas that could be compressed if there were no gas expansion. The symbol " V_d " is commonly used to express this volume. It depends on piston size, speed and length of stroke, and whether the piston is single- or double-acting.

The calculation of engine power must be obtained from the energy required by the gas per stage. Because of the gas expansion in the cylinder discussed above, as this ratio goes up, engine power increases.

The ratio (V_1/V_d) is known as *volumetric efficiency*. It decreases with increasing compression ratio and an increase in volume V_c . Thus, compression ratio is an economic consideration.

Compression Ratio Per Stage

The compression ratio per stage (R) seldom exceeds 6:1. The volumetric efficiency declines, discharge temperature rises, and mechanical stress limitations become more pronounced as R increases. In actual practice, R seldom exceeds 4:1 when boosting gas from low pressure for processing or sale. When the total pressure ratio is greater than this, multiple stages of compression are used.

It may be shown that the total power is a minimum when the ratio in each stage is the same. This may be expressed in equation form as

$$R = \left(\frac{P_D}{P_1} \right)^{1/n} \quad (15.20)$$

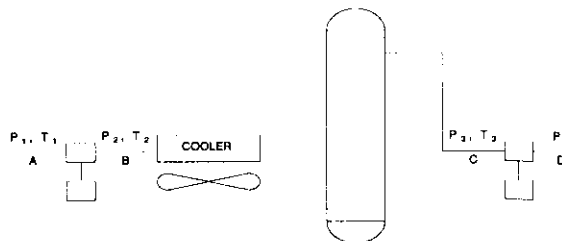
Where: P_D = final discharge pressure
 P_1 = suction pressure
 n = number of stages required

Where several stages of compression are used, it is necessary to have an interstage cooler and scrubber to cool the gas from the first stage and then collect any liquids formed on cooling. This interstage pressure drop should seldom exceed 35 kPa [5 psi]. In order to account for this drop, the R found in Equation 15.20 is usually corrected. One common method is to assign half of the pressure drop to each stage. Sometimes the compression ratio is reduced in the higher pressure stages because of rod loading limitations.

The R from Equation 15.20 is a theoretical one, for it makes no provision for the pressure drop through the interstage piping, cooler, and separator.

The schematic diagram below shows the typical layout for a two-stage compression plant. The general procedure for finding engine horsepower is as follows:

1. Calculate R from Equation 15.20 and multiply by P_1 .
2. Estimate pressure drop, $P_2 - P_3$.
3. Actual $P_2 = R(P_1) + 1/2$ of pressure drop in Step 2.
4. Actual $P_3 = P_2 - \Delta P$ between stages, using P_2 from Step 3.
5. Estimate T_3 , the suction temperature to the second stage. (Ambient dry bulb plus 15°C for air cooling or wet bulb plus 13°C for water cooling.)
6. Calculate actual " R " for each stage, calculate power for each stage, and add to get total engine power needed.
7. The total heat load in the cooler is the sum of the gas sensible heat from T_2 to T_3 , and the total latent heat of all fluids condensed (water plus oil).



Once these pressures have been determined, it is possible to calculate theoretical work. This can be corrected with an appropriate efficiency to estimate actual work.

Overall Efficiency - "E"

As previously discussed, the overall efficiency is a function of compressor design details, suction pressure, speed, loading, compression ratio, and general mechanical condition of the unit. For detailed information, the vendor should be contacted. It is, however, possible to estimate efficiency for general planning purposes from general correlations as shown in Figures 15.12 and 15.13 .

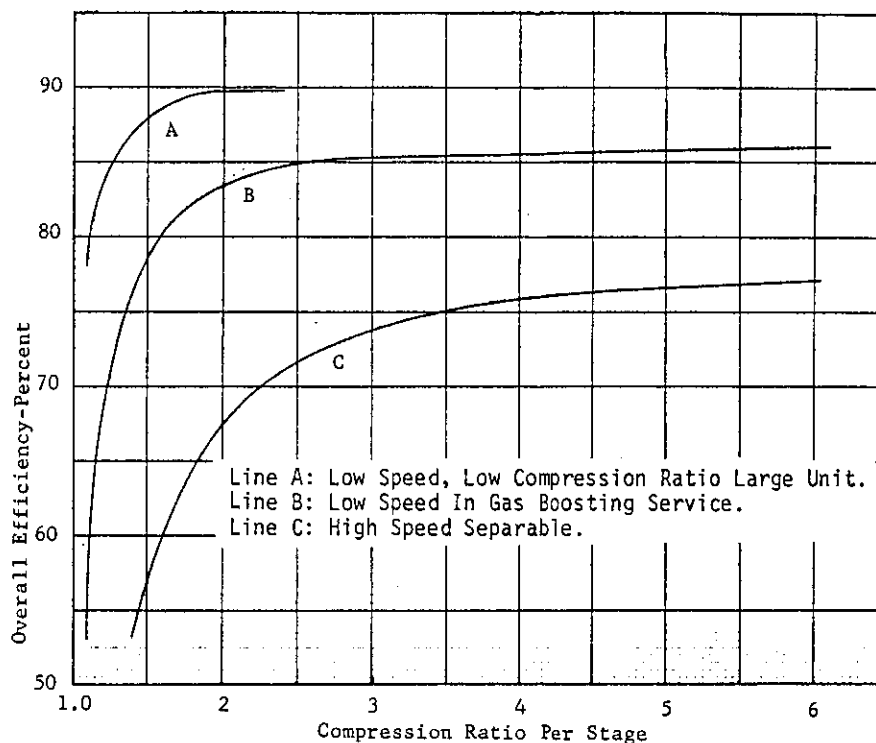


Figure 15.12 Overall Efficiency for Reciprocating Compressors with Air

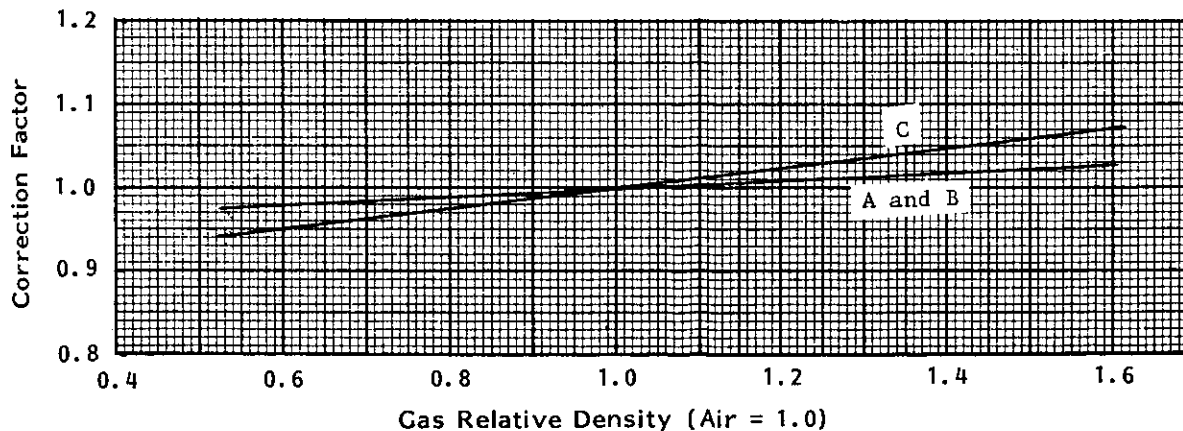


Figure 15.13 Gas Relative Density Correction for Figure 15.12

Figure 15.12 is for air compression. Figure 15.13 is a correction factor correlation for gas relative density that is multiplied by the efficiency of Figure 15.12. Curve A is for large integral or separable units, at low compression ratios, typical of large pipeline operations. Curve B is typical of large integral or separable units at speeds of less than 600 rpm. Curve C is for small separables operating generally above 900 rpm. Many units operate between the curves shown. Curves like these are for planning purposes only.

The values shown normally will apply at suction pressures greater than 200 kPa (abs.). For suction pressures lower than this, the efficiency may be 15-20% lower than shown. The compression ratio shown is per stage.

Example 15.2: 280 000 std m³/d [9.9 MMscfd] of a 20.3 MW gas is compressed from 700-6000 kPa [101-870 psia] in a low speed integral compressor. First and second stage suction temperatures are 40°C [104°F] and 45°C [113°F] respectively. Interstage pressure drop is 40 kPa [6 psi]. Estimate the power requirements.

1. Assume equal compression ratios per stage and calculate the compression ratio including the interstage pressure drop.

$$\text{From Equation 15.20, } R = \left(\frac{6000}{700} \right)^{1/2} = 2.93$$

Include interstage pressure drop

$$P_1 \text{ (1st stage)} = (2.93)(700) + (0.5)(40) = 2069 \text{ kPa}$$

$$P_1 \text{ (2nd stage)} = 2069 - 40 = 2029 \text{ kPa}$$

$$R_1 = (2069/700) = 2.96$$

$$R_2 = (6000/2029) = 2.96$$

2. From Figure 15.13, $E = (0.852)(0.982) = 0.837$

From Figure 15.3,

$$\text{1st stage } z_1 = 1.0, \quad z_D = 0.98$$

$$\text{2nd stage } z_1 = 0.955, \quad z_D = 0.945$$

Assume $k = 1.25$, so $m = 0.2$

- 3(a). Calculate power using Equation 15.5

$$\text{1st stage } \text{kW} = \left(\frac{11.57}{0.837} \right) \left(\frac{1.25}{0.25} \right) (0.28) \left(\frac{100}{288} \right) (313)(2.96^{0.2} - 1)(0.99) = 505$$

$$\text{2nd stage } \text{kW} = \left(\frac{11.57}{0.837} \right) \left(\frac{1.25}{0.25} \right) (0.28) \left(\frac{100}{288} \right) (318)(2.96^{0.2} - 1)(0.95) = 492$$

- 3(b). Calculate power from Figure 15.2(a)

$$\text{1st stage } @ T_1 = 40^\circ\text{C}, P_1 = 700 \text{ kPa} \quad h_1 = 1560 \text{ kJ/kmol}$$

$$@ S_D = S_1 \text{ and } P_D = 2069 \text{ kPa} \quad h_{2_{\text{isen}}} = 4600 \text{ kJ/kmol}$$

$$\text{kW} = \frac{(0.28)(1739)(4600 - 1560)}{(0.837)(3600)} = 491 \text{ kW}$$

$$\text{2nd stage } @ T_1 = 45^\circ\text{C}, P_1 = 2029 \text{ kPa} \quad h_1 = 1570 \text{ kJ/kmol}$$

$$@ S_D = S_1 \text{ and } P_D = 6000 \text{ kPa} \quad h_{2_{\text{isen}}} = 4460 \text{ kJ/kmol}$$

$$\text{kW} = \frac{(0.28)(1739)(4460 - 1570)}{(0.837)(3600)} = 467 \text{ kW}$$

Example 15.2 (Cont'd.)

3(c). Calculate power from Figure 15.4

$$@ k = 1.25 \text{ and } R = 2.96, \text{ Power} = \frac{172 \text{ kW}}{10 \times 10^5 \text{ std m}^3/\text{d}}$$

$$\text{1st stage} \quad q = \left(\frac{280\,000}{100\,000} \right) \left(\frac{100}{100} \right) \left(\frac{313}{288} \right) = 3.04$$

$$\text{Power} = (172)(3.04) \left(\frac{0.82}{0.837} \right) = \underline{513 \text{ kW}}$$

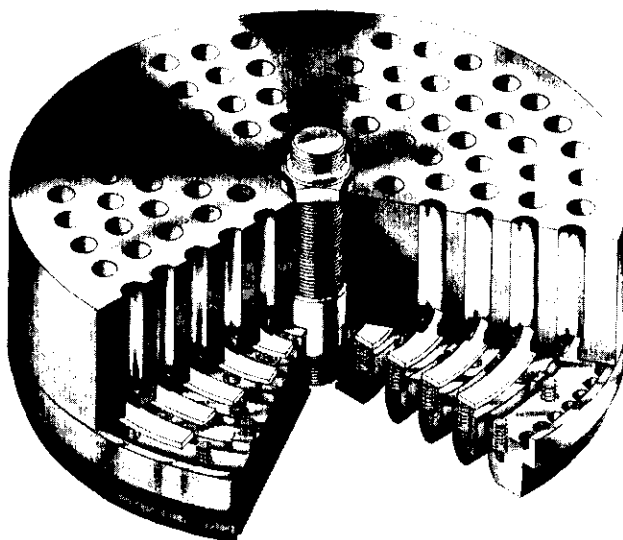
$$\text{2nd stage} \quad q = \left(\frac{280\,000}{100\,000} \right) \left(\frac{100}{100} \right) \left(\frac{318}{288} \right) = 3.09$$

$$\text{Power} = (172)(3.09) \left(\frac{0.82}{0.837} \right) = \underline{521 \text{ kW}}$$

The range of values obtained in Example 15.2 is normal. The deviation between the three methods probably is less than the errors in the data being used in the methods.

What do we have at this point? The power that must be transmitted to the compressor to satisfy the gas energy needs, with allowance for efficiency. This must be supplied by a driver (engine) capable of producing this necessary shaft work. Drivers will be discussed in a separate section because different types may be used with a given compressor.

The value of efficiency (E) depends on the type of compressor and the model within a given type. It includes both a correction for the isentropic assumption and the actual mechanical efficiency of the equipment.



Reciprocating Compressor Valve

Courtesy Cooper Industries

Volumetric Efficiency

In Figure 15.10 the general behavior of a reciprocating cylinder is shown. The volume of new gas entering the cylinder (BC) is less than the maximum amount of gas that could enter (ABC) because of expansion of gas trapped between the piston and the end of the cylinder. This maximum volume is called *displacement* (V_d). The ratio of actual volume entering (V_1) to V_d is called *volumetric efficiency*.

The equation relating the variables affecting volumetric efficiency is

$$E_{vz} = \frac{V_1}{V_d} = 0.96 + (C - L) - (C)(R)^{1/k} \left(\frac{Z_1}{Z_D} \right) \quad (15.21)$$

Where: E_{vz} = volumetric efficiency
 $C = (V_c/V_d)$ = clearance, expressed as a fraction of displacement volume
 R = compression ratio
 V_1 = volume of gas per unit time at actual suction conditions
 V_d = piston displacement per unit time
 $L = R/100$, where R = compression ratio/stage at speeds of 200-500 rpm
 $= R/50$, where R = compression ratio/stage at speeds above 500 rpm

The value of clearance (C) in Equation 15.21 is expressed as a fraction. As noted in Figure 15.10, it is clearance volume (V_c) divided by piston displacement (V_d). This number may be varied at will.

The number 0.96 in Equation 15.21 is to account for any loss of volumetric efficiency due to valves and ports, losses through the piston rod packing, etc. This number represents a practical average and may be as low as 0.93 for high speed units using lubricated rings and less than 0.9 for non-lubricated cylinders.

Volume " V_1 " is the flow rate of gas to the suction valve of a given stage at the actual P_1 and T_1 of the point in the system.

$$V_1 = V_s \left(\frac{P_s}{P_1} \right) \left(\frac{T_1}{T_s} \right) (Z_1) \quad (15.22)$$

Where: V_1 = volume/min of gas at P_1 and T_1 (suction conditions)
 V_s = vol. rate per min. of flow at standard P_s and T_s
 Z_1 = compressibility factor at P_1 and T_1

Displacement volume is dependent on piston area, stroke, speed and whether the cylinder is single or double acting.

$$V_d = (A)(d^2)(\text{Stroke})(\text{rpm})(\text{Factor}) \quad (15.23)$$

Where:		Metric	English
		m^3/min	ft^3/min
	V_d = displacement volume/min		
	A = conversion factor	7.85×10^{-7}	0.000 454
	Stroke = piston stroke	cm	in.
	d = cylinder diameter	cm	in.
	Factor = 1.0 for single acting cylinders and slightly less than 2.0 for double acting (to account for piston rod volume). A value of 1.9 is often used for planning purposes		
	rpm = revolutions of driving shaft per minute		

Equation 15.21 is an accurate representation of cylinder performance. If V_1 , the gas available for compression changes, the value of the right hand side must change to preserve the equality shown. If a control is not used to keep the equation balanced, R will change as necessary to do so. Since discharge pressure is often fixed, this means that suction pressure varies automatically. The control of compressors will be discussed in several later sections.

Equation 15.21 can be used to size cylinders. With all other variables fixed, it can be solved for V_d . One then solves Equation 15.23 for d^2 and thus d . The allowable piston diameter is limited by frame, rod and pin loading specified by the manufacturer.

Example 15.3: Using the values from the previous example estimate the cylinder sizes for the following compressor.

Speed = 400 rpm

Stroke = 21.6 cm [8.5 in]

Normal clearance:

1st stage = 7%

2nd stage = 12%

Assume 2 cylinders per stage and all cylinders are double-acting

1st stage – 1. Estimate E_{vz} from Equation 15.21

$$E_{vz} = 0.96 + (0.07 - 0.0296) - 0.07 (2.96)^{0.8} \left(\frac{1.0}{0.98} \right) = 0.830$$

2. Determine V_1 from Equation 15.22

$$V_1 = \left(\frac{280\,000}{1440} \right) \left(\frac{100}{700} \right) \left(\frac{313}{288} \right) (1.0) = 30.19 \text{ m}^3/\text{min}$$

$$= 15.09 \text{ m}^3/\text{min per cylinder}$$

3. Calculate d from Equation 15.23

$$V_d = \frac{V_1}{E_{vz}} = \frac{15.09}{0.830} = (7.85 \times 10^{-7})(d^2)(21.6)(400)(1.9)$$

Solving for d ; $d = 37.6 \text{ cm}$

2nd stage – 1. $E_{vz} = 0.96 + (0.12 - 0.0296) - 0.12 (2.96)^{0.8} \left(\frac{0.955}{0.945} \right) = 0.0762$

$$2. V_1 = \left(\frac{280\,000}{1440} \right) \left(\frac{100}{2029} \right) \left(\frac{318}{288} \right) (0.995) = 10.11 \text{ m}^3/\text{min}$$

$$= 5.05 \text{ m}^3/\text{min per cylinder}$$

$$3. V_d = \frac{V_1}{E_{vz}} = \frac{5.05}{0.762} = (7.85 \times 10^{-7})(d^2)(21.6)(400)(1.9)$$

$d = 22.7 \text{ cm}$

Rod, Pin, and Frame Loadings

Whatever the name used, this refers to the maximum stress that may be placed on the "weakest link" in the drive mechanism. This will vary with the machine. It may be the rod, the crosshead pin or bushing. This may be different in compression and tension.

The following equations apply:

$$\text{Compression} \quad L_c = (A_h)(P_2) - (A_c)(P_1) \quad (15.24)$$

$$\text{Tension} \quad L_t = (A_c)(P_2) - (A_h)(P_1)$$

Where: L = load (in force units)

A_h = area of piston at head end

A_c = effective area at crank end (area of piston - area of rod)

P_1 = suction pressure

P_2 = discharge pressure

In any service, it is critical that these loads not be exceeded. Allowable rod loads are available from the compressor manufacturer.

DRIVERS (ENGINES)

Once the compressor power needs have been established, it is necessary to choose a driver to supply that amount of power. This driver may be some kind of an engine, gas turbine, electric motor or an expansion turbine. The choice depends on the compatibility of compressor and driver, fuel availability, weight and volume limitations, etc. If other factors are not critical, compatibility is most important. Comparable speed range, for example, is one consideration. One can use speed increasers and speed reducers satisfactorily but they should be avoided unless their use is better than the other alternatives.

All drivers have a rated capacity. But, this may not be the amount of power that can be produced reliably and continuously under the conditions present. Some adjustment or derating may be necessary. All combustion engines, for example, have an output power which depends to some degree on combustion air density. Both altitude and temperature affect output.

Some of the available engine power may be used to drive accessories such as pumps and coolers. This amount of accessory power, if any, must be provided for in sizing the engine.

On a platform the accessories may be powered from an independent "power package" used for general purposes. Within a frontier area or an isolated unit, the driver may have to power all accessories. In some cases, lights and other noncompressor auxiliary needs are powered by the driver. In the early planning before such details are known, it is wise to be generous in estimating power requirements.

Reciprocating Engines

This type of engine comes in many sizes and speeds. In large sizes, up to 600 rpm, the compressor and engine may be integral units. They share the same crankshaft. Up to about 150 kW, use of V-belts to connect the two may be employed. An intermediate type may connect the end of the engine crankshaft to the compressor – directly or through a speed controller.

Rated Power

The engine supplied will be rated according to a somewhat arbitrary system. An engine (driver) furnishing power by combustion of fuel in reciprocating cylinders will use a rating involving the following:

1. A derating of test-stand power for high speed, low power engines
2. Altitude corrections
3. Use of engine power to drive accessories like water and lube oil pumps, fan for aerial cooling, etc.

Engine Derating. - Small compressor units often use high speed engines, using a rating based on conditions which cannot be maintained reliably in actual service. This is sometimes called a test-stand power rating – a maximum power output under ideal conditions, for short periods of time.

Many approaches are used to derate this ideal power in order to obtain a reliable, low-maintenance operation. These include:

1. Derating test-stand power by 35-45%
2. Limiting BMEP
3. Limiting rpm
4. Limiting piston speeds (it is common to limit compressor cylinder piston speeds to about 275 m/min and power cylinder piston speeds to 300 m/min)

In many companies, all of these are applied to see which limits the desirable BHP output of the engine.

Integral type units, where the power and compressor cylinders are mounted on the same frame, normally are not derated by anything except BMEP.

Any derating done is a compromise economically. You are buying more potential power which will return the extra capital cost through decreased cost for maintenance and less loss of revenue because of downtime. This is necessarily a judgment call based on experience.

Altitude Correction. - Altitude affects the density of the air available for combustion. Any rating must be based to some degree on this density.

The effect of density depends on the engine. There are two basic types:

1. Nonsupercharged or nonturbocharged (naturally aspirated) engines where the air is burned at its natural pressure
2. Supercharged or turbocharged engines where the air is compressed before combustion.

The altitude deration varies with the manufacturer and engine type. The following summary is typical:

High Speed

Nonturbocharged - 10% for each 1000 m [3300 ft] above sea level

Turbocharged - 6.5% for each 1000 m [3300 ft] above sea level

Larger Integral Units

No correction up to 500-800 m [1650-2600 ft] above sea level, then 10% per 1000 m [3300 ft] for natural aspirated and 6.5% per 1000 m [3300 ft] for turbocharged

(Some large engines are designed for no correction up to 2100 m [6900 ft] above sea level)

The actual altitude deration will be specified by the manufacturer.

Temperature Correction. - Temperature also affects air density. Temperature corrections vary with engine type and manufacturer. Deration of 1-2% per 5.5°C [10°F] is typical for many engines. Actual temperature factors can be obtained from manufacturer.

Accessories Correction. - If the compressor engine drives all of its accessories, not all of the horsepower output is available to compress the gas. The exact amount of accessory horsepower is dependent on the engine. For a naturally aspirated engine, accessories will use about 3-5% of the engine power.

The following formula is used to correct for altitude and accessories:

$$\text{Rated kW} = \frac{\text{kW}}{(1 - \text{Acc. Corr.} - \text{Alt. Corr.} - \text{Temp. Corr.})} \quad (15.25)$$

Where: kW = power required by the compressor

Acc. Corr. = that fraction of the engine power used for accessories

Alt. Corr. = fractional derating for altitude effects

Temp. Corr. = fractional deration for temperature effects

Note: In addition to this correction, one still must consider BMEP and any speed limitations.

Specification of Reciprocating Engines

The choice is a function of engine, rpm, weight, speed and the like because they all affect cost and performance. The following must be considered.

1. Brake Mean Effective Pressure (BMEP) -- the theoretical average pressure needed in the power cylinder throughout the power stroke to develop the rated power.

$$\text{BMEP} = \frac{(A)(\text{kW})}{(\text{no. cycles})(d^2)(L)(N)} \quad (15.26)$$

		Metric	English
Where:	BMEP = in units of	MPa	psia
	L = length of stroke	cm	in.
	d = power cylinder diameter	cm	in.
	A = conversion factor	77 780	677 200
	N = rpm for two cycle engines = (rpm/2) for four-cycle engines		
	kW = engine power output requirements		

Four-cycle engines must develop a higher BMEP at the same speed, for a give load, due to the additional crankshaft revolution taken per power stroke. Per unit weight, a four-cycle engine must operate at a higher BMEP, rpm, or both, to produce the same amount of power.

Maintenance is dependent to some extent on BMEP. Individual experiences vary, but the following are reasonable limits to produce minimum maintenance overhead:

- BMEP less than 0.55 MPa [80 psia] for high compression heads
- BMEP less than 0.45 MPa [65 psia] for low to medium heads
- BMEP less than 0.86 MPa [125 psia] for supercharged engines.

These limits depend to some extent on the heating value of the fuel

2. Piston speeds for compressor cylinders should generally be limited to 275 m per min [900 ft/min]. Power piston speed limits of 300 m per min [985 ft/min] are most common.
3. Set up derating criteria suitable for the types of engines being considered. This may include a mere power deration for high-speed engines. Some use a specification limiting power output per unit weight of engine.
4. Cooling – if inter-coolers and/or after-coolers are needed, specify the approach desired.
5. Controls – these should be specified, even though the control package is fairly standard.
6. Piston loading – this is normally specified by the vendor but should be considered by all parties. Operating near the maximum allowable loading may be a future source of trouble.
7. Fuel – specify the quality available (the lower the heating value the better). Gas with a heating value above 54 000 kJ/std m³ [1450 Btu/scf] is basically undesirable although sometimes necessary. Avoid use of high heating value gases if at all feasible, particularly gases with substantial quantities of C₅+ fractions.

Each of the above factors is open for individual interpretation. Regardless of this, they must always receive some degree of formal consideration.

The choice of supercharged versus naturally aspirated engines is likewise somewhat arbitrary. The former have a lower fuel consumption, are best at high elevations, have a low cost per horsepower, and may require less cooling. Naturally aspirated engines are usually less susceptible to problems associated with high ambient temperatures, are easier to operate unattended, and may operate better on high Btu fuel. They inherently have less overload capacity than supercharged engines.

Gas Turbines

The gas turbine is used where high horsepower output per unit weight is desirable. The capital cost is favorable, although the fuel efficiency is frequently less than a reciprocating engine. They are very compatible in performance characteristics with centrifugal and axial compressors and are often used to drive them.

The basic mechanical and thermodynamic cycles are shown in Figure 15.14. Part (a) of this figure shows a simple open cycle. Gas and excess air are burned in the combustion chamber. A portion of the energy produced is available for net work. The remainder drives the inlet air compressor.

What Happens in a Gas Turbine

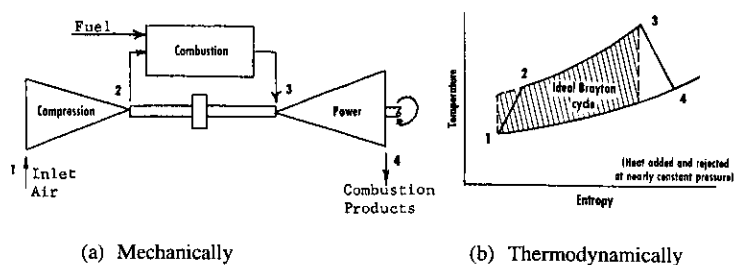


Figure 15.14 Schematic of Simple Turbine Operation

The ideal cycle shown in Figure 15.14 (b) is the Brayton Cycle. Points 1-2-3-4 in that figure show the actual cycle. This ideal cycle may be used to outline pertinent characteristics of a turbine. Figure 15.15 will be used for this purpose. The air enters at Point 1 and is compressed isentropically to temperature T_2 and pressure P_2 . Steps 2-3 (combustion) take place at substantially constant pressure. Step 3-4 is an isentropic expansion through the power turbine. In an actual turbine, Steps 1-2 and 3-4 will not be isentropic.

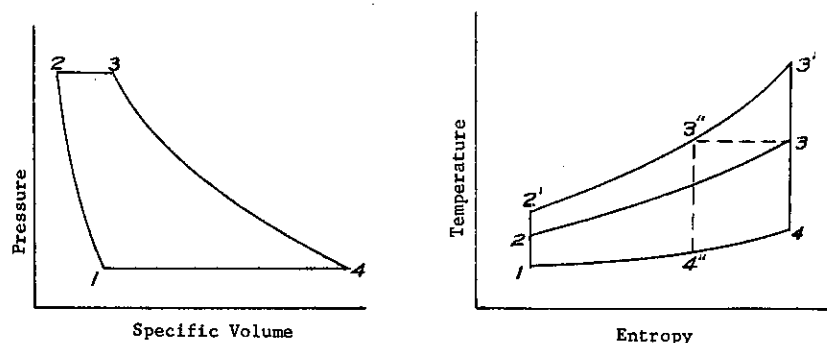


Figure 15.15 Thermodynamic Diagrams for Ideal Brayton Cycle

A gas turbine is a thermally rated device. The thermal efficiency used must be a function of pressure ratio, according to Figure 15.15. For the ideal cycle,

$$E_{\text{theor}} = 1 - \left(\frac{T_1}{T_2} \right) = 1 - \left(\frac{1}{(P_2/P_1)^m} \right) = 1 - \left(\frac{T_4}{T_3} \right) \quad (15.27)$$

Where: E_{theor} = thermal efficiency
 $m = (k - 1)/k$

Note that this thermal efficiency should not be confused with the overall efficiency used previously for compressors.

The overall cycle efficiency is given by

$$E_{\text{overall}} = \frac{W_a}{(m_f)(\text{LHV})} \quad (15.28)$$

Where: W_a = actual shaft work at power turbine, kW
 m_f = mass flow rate of fuel, kg/s
 LHV = lower heating value of fuel, kJ/kg

The thermal efficiency of a gas turbine will decrease about 1.1% for each 10% the power output is reduced below rated output.

If one increases pressure ratio, the cycle will be 1-2'-3'-4 instead of 1-2-3-4. In an actual turbine, the maximum temperature (T_3) is fixed by metallurgical limitations. If T_3 is the allowable maximum temperature, the theoretical cycle will be 1-2-3''-4''. Thermal efficiency is, therefore, limited by both pressure ratio and temperature. As the pressure ratio increases for Step 1-2, more power is needed for air compression. The most efficient gas turbines available currently employ compression ratios of 20:1 and combustion temperatures in excess of 1200°C [2200°F].

Metallurgical advances which will enable temperature to be raised will have dramatic effects on efficiency. Excess air of 200-300% is commonly used to keep this temperature within bounds. The "backwork" necessary to drive the air compressor is therefore large. It might require from 40-80% of the power turbine output. The Saturn turbine, for example, produces about 3000 hp in the power turbine to produce about 1100 hp in net shaft work available for productive use.

Figure 15.16 is a drawing of a turbine driving a pipeline compressor. This is an older model unit but it illustrates the key components in the system. The most critical temperature in the turbine is that entering the high pressure turbine. Current models may run at higher temperatures than shown.

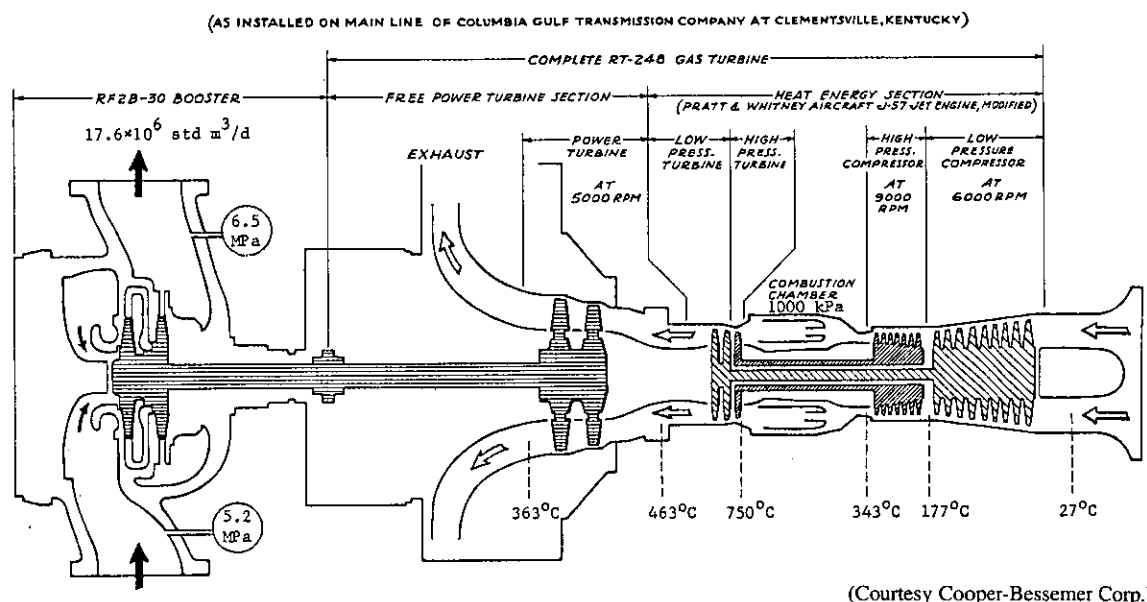
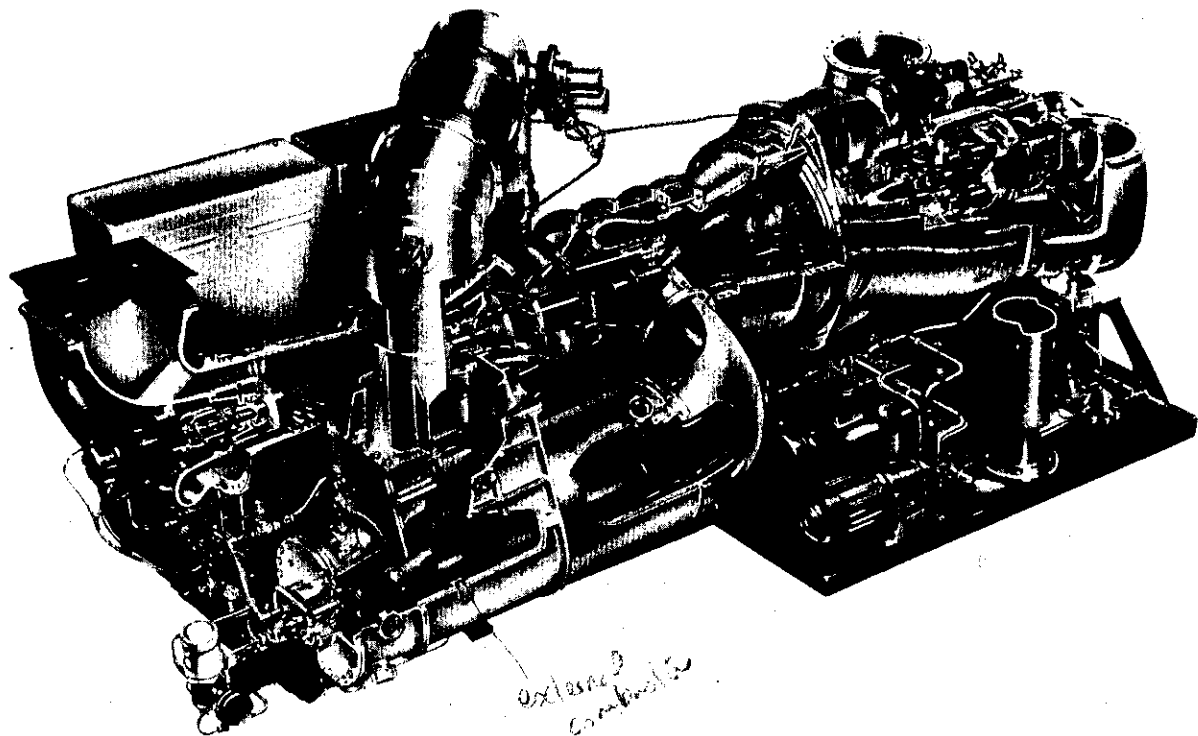


Figure 15.16 Cutaway View of a Gas Turbine

Current designs employ combustion temperatures of 1200°C [2200°F]. Gas turbine efficiencies vary from 22% for older models to near 40% for state-of-the-art designs.

Figure 15.16 tends to oversimplify the complexity of the turbine system. The drawing on the next page is a cut-away view of an industrial turbine. Although simple in principle, the turbine involves the proper maintenance of many parts, some operating under severe service conditions.

Two types of gas turbines have historically been supplied to the oil and gas industry. These are industrial and aircraft derivative, although the distinction between the two has narrowed in recent years. Industrial turbine typically employ heavy horizontally split casings are designed for long life and long run times between maintenance. They are more durable and tolerate a wider range of fuels. Disadvantages include weight and need for on-site maintenance. Aircraft derivative turbines are lighter and more compact. They employ modular construction which minimizes downtime for maintenance and repairs.



Aircraft derivative engines consist of an air compressor, combustor and high pressure turbine used to supply power to the compressor. They are often called gas generators because they supply hot, medium pressure exhaust gas to a power turbine which drives the driver equipment. The power turbine is frequently supplied by a different manufacturer than the hot gas generator. When the power turbine is separate from the high pressure, gas generator turbine the unit is called a split shaft or two shaft machine. With the two-shaft turbine it is possible to operate the driver at a different rpm than the gas compressor. Several gas turbine configurations are shown in below Figure 15.17.

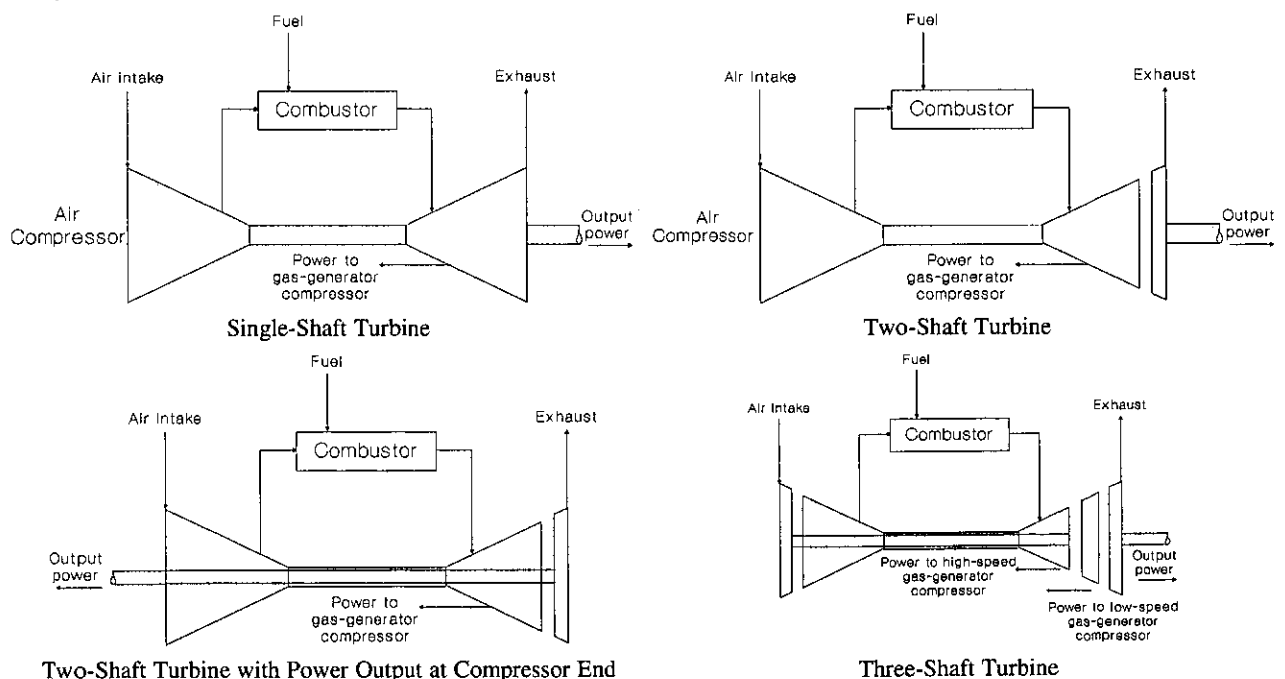
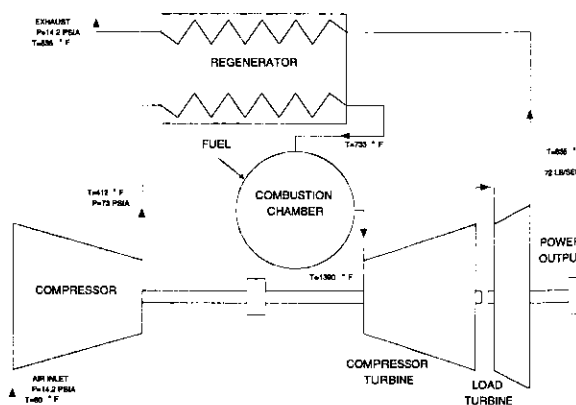


Figure 15.17 Gas Turbine Configurations

When choosing between single and two-shaft machines, one must compare their operating characteristics with those of the system involved. It must be remembered that flexibility is one requisite. Two shaft machines are preferred for gas compression service because of the ability to vary speed and maintain maximum power output. Single shaft machines are often used in power generation because of the constant speed requirement and the higher inertia of the rotating system.

Regenerators. - The total thermal efficiency of the system is sometimes improved by using a regenerator, as shown in the figure to the right. The conditions shown are for an open cycle, two-shaft machine with 10 stages in the axial compressor and two turbine stages. The regenerator is designed for 75% effectiveness. This regenerator preheats the air. The exhaust air is still quite hot and may be used for subsequent heating by simply exhausting at a slightly higher pressure.

The thermal efficiency for a simple gas turbine may vary from 20-25% without regenerator and from 25-28% with a regenerator. The regenerator exhaust gases usually contain about 17-18% oxygen. A back-pressure up to about 200 kPa may be carried on the regenerator. Regenerators are not feasible on today's high efficiency turbines because there is little temperature difference between the turbine exhaust and the air compressor discharge.



Schematic of a Split-Shaft Turbine with Regenerator

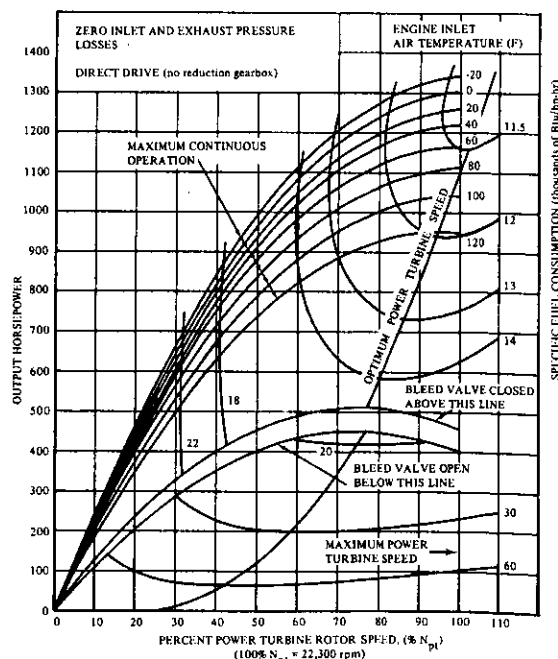
Complex Cycles. - One advantage of a gas turbine is its compatibility with other energy components. For example, a gas turbine and expansion turbine may be directly coupled with one governing system controlling the tandem unit. When a plant requires both power and heat, a gas turbine used with a waste-heat boiler might give an overall cycle efficiency of 55-80%, depending partially on whether or not the boiler is unfired or fully fired. Fuel needs chargeable to power vary from about 4000-10 000 kJ per kW-h [2830-7070 Btu/hp-hr]. These combined-cycle units are frequently used to generate electricity and process steam at the same time.

Performance. - The curve shown at right is for a single-shaft Solar turbine operating at sea level. Each turbine possesses its own unique characteristics but the curve shown is typical. This turbine provides maximum power output at 22 300 rpm.

The ISO (International Standards Organization) rated power output of a gas turbine as the output at design speed, sea level, 15°C ambient temperature and zero inlet and exhaust pressure drops.

The site output of any turbine is governed by altitude and temperature. The ISO has set up standard derating criteria for these factors, at a given speed. If the power rating is established at sea level, deduct about 1.1% for each 100 m [330 ft] the site is above sea level. This altitude deration factor is a straight line and can be applied to any reference altitude.

Temperature of the entering air is a critical factor affecting power output. The ISO temperature deration line is also linear. Power output should be



derated about 0.9% for each 1°C that the temperature rises above the rating temperature, which commonly is 15°C [59°F]. If the temperature is lower than this, the rating is increased by the same amount.

In arctic conditions, where very low temperatures are encountered, the potential power output exceeds rated output. However, there are special problems at extremely low temperatures.

Site derating for inlet and exhaust duct losses varies from turbine to turbine but some typical values are 0.2% per mbar for the inlet and 0.1% per mbar for the outlet. Note: 1" H₂O ≈ 2.5 mbar.

Normal pressure drops for inlet and exhaust ducts range from 5-15 mbar [2-7.5 in H₂O]. Waste heat recovery in the exhaust can significantly increase pressure drop. these losses must be accounted for when site rating a turbine.

The factors affecting turbine speed are the same as those for centrifugal and axial compressors. The front end of a turbine is an axial air compressor.

Environmental Impact

Environmental control of a gas-turbine installation in a process plant is governed by local regulations plus company practices at the plantsite. The environmental impact falls into two distinct area, noise and stack emissions.

To meet the normal noise criterion of 85 to 90 dBA at 1 meter distance, inlet and exhaust silencers together with acoustic enclosures are required. The methods and components used for noise abatement are identical, regardless of the type of gas turbine used, to a point where one can talk about near-standard procedures.

In step with increasing demands for noise control the design and construction of the silencers and acoustic enclosures have undergone a quiet evolution. The noise-abatement components are now of heavy structural design and often of the free-standing type. Outer walls may be fabricated of 3/16 to 1/4-in.-thick steel plates, and the materials for the acoustic walls may be heavy-gage galvanized-steel perforated plate for the air intake and enclosures, unless special circumstances dictate otherwise. High-temperature corrosion-resistant materials are used for the exhaust stack. The sound-absorbing materials may be mineral or glass fiber.

The operational efforts required for stack-emission control for a process-type gas-turbine installation will normally be concerned only with the amount of oxides of nitrogen, NO_x, in the exhaust. This is because the strict turbine fuel specifications, as a side-effect, eliminate most polluting agents at the source, i.e., from the fuel before it is burned. Normal stack pollutants, such as sulfur dioxide, carbon monoxide, and particulates, are controlled by the fuel specifications and the turbine's combustor design, and do not pose an operational problem. Unfortunately, this is not the case with NO_x.

Formation of NO_x is complex and not a subject for this writing. However, in simple terms, NO_x production for a gas turbine is influenced by ambient conditions – it decreases with increasing relative humidity and increases with flame temperature and residence time at this temperature. Different fuels produce different levels of NO_x; this again has to do with flame temperature: some low-Btu fuels, e.g., hydrogen, may have high NO_x-emission levels.

The NO_x emission from a gas turbine can be controlled by injecting water of boiler-feed quality into the combustors. In this way, flame temperature can be reduced and NO_x emission held at acceptable levels as dictated by site location and local regulations.

Low-NO_x combustors can also reduce NO_x emissions. These are sometimes called "lean burn" combustors and are designed to lower peak flame temperatures.

One of the most effective NO_x-control techniques is selective catalytic reduction (SCR), in which anhydrous or aqueous ammonia is injected into the fluegas upstream of a catalyst bed. The NO_x and NH₃

combine at the catalyst surface, forming an ammonium salt intermediate that subsequently decomposes to produce elemental nitrogen and water.

In SCR, the NO_x -reduction reactions occur only in a narrow temperature range, generally 260-500°C [500-932°F], depending on the type of catalyst. Operation above the maximum temperature results in oxidation of ammonia to either NO_x or ammonium nitrate and ammonium nitrite; operation below the optimum range does not provide the energy necessary to initiate the reaction. Figure 15.18 shows a schematic of an SCR system.

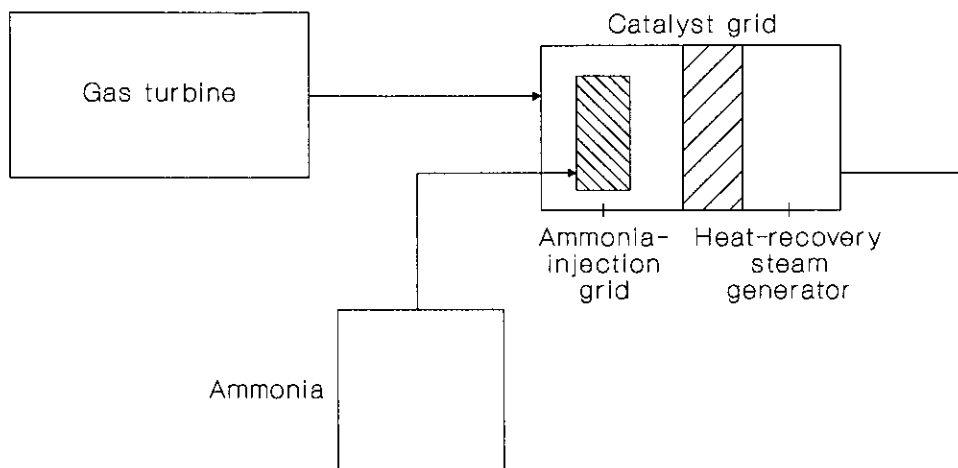


Figure 15.18 Selective Catalytic Reduction for NO_x Emission Control on a Gas Turbine Exhaust

Fuel and air quality

Clean fuel and inlet air are absolute necessities if one is to expect trouble-free operation and life. Gas turbines run on a variety of fuels (crude oil, NGL, diesel, natural gas, etc.). In production operations, natural gas is the most common fuel. When natural gas is used it is imperative that liquid hydrocarbons not enter the turbine. If available, dry, processed gas should always be used as fuel, if not, the fuel gas should be pre-heated well above its hydrocarbon dew point.

A gas turbine ingests massive amounts of air. The air rate to a G.E. Frame 3 [10 000 kW] turbine is approximately 52 kg/s [114 lbm/sec]. Atmospheric pollutants can cause fouling and corrosion to turbine components. Salt water spray, drilling need, cement, sandblast are the most common contaminants in off-shore service. A properly designed air filter is imperative. It should be designed for easy maintenance and low pressure drop. Rain hoods and de-icing equipment should also be installed if dictated by climate.

Exhaust ducts should be oriented so as not to interfere with other process operations. They should be installed so that the prevailing wind does not carry the hot air plume into fin-fan coolers, air conditioning systems, heliports, other engine intakes, etc.

Expansion Turbines

This form of driver will be discussed in more detail in Chapter 16, Refrigeration. An expander must be driving something to produce the refrigeration but it usually is chosen primarily for its refrigeration potential.

Electric Motors

Electric motors are probably the simplest driver used for compressors and pumps. They have a high efficiency (the electrical power generation plant may not!), are rugged, reliable, and require little maintenance. In larger sizes (> 100-200 kW) the manufacturer will build the motor exactly to specification.

In compressor service, the primary disadvantage of electric motor drivers is their constant speed. Variable speed motors are available but are economical only in the smaller sizes. A second disadvantage is starting torque. If, during start-up, the motor torque is less than that required by the driven equipment the motor will "stall," and the speed drops rapidly to zero.

The best way to evaluate torque requirements is to superimpose the speed versus torque curve of the motor; these curves are provided by the respective manufacturers. NEMA has established minimum torque requirements for three basic motor designs: Design B, for normal centrifugal loads; Design C, for loads requiring high starting torque (reciprocating compressors); and Design D, for high-slip high-inertia loads such as flywheel drives on machine tools.

Starting current is also a problem. A motor usually draws 5-7 times its full load current when starting. This may depress voltage and depresses starting torque in proportion to the voltage drop squared. This is not a problem if the reduced torque is still sufficient to turn the compressor.

Induction motors operating at 1200, 2400, or 3600 rpm (60 Hz) are the most popular driver to about 1-2 MW [1300-2600 hp]. Synchronous motors are often preferred in the larger sizes. Synchronous motors are often lower in installed cost, more efficient, and have a higher power factor.

In the United States, the maximum synchronous speed of a motor will be 3600 rpm [3000 rpm at 50 Hz]. Induction motors will operate slightly below the synchronous speed due to slip. Increasing the number of poles, or pairs of poles, will lower the synchronous speed. Speeds are available at 1800, 900 and 450 rpm. Low speed synchronous motors are frequently used to drive reciprocating compressors.

Motor Enclosures

There are several types of motor enclosures.

1. Open drip-proof – for use indoors in nonhazardous locations
2. Weather Protected II – for use outdoors in nonhazardous locations
3. Totally enclosed, water-air-cooled – for use indoors or outdoors in clean or dirty, nonhazardous locations
4. Totally enclosed, fan-cooled – an alternative for nonhazardous locations
5. Hazardous locations – where flammable vapors are present

A Class I environment contains flammable gas or vapor. Within this class, several groups have been specified. Most hazardous petroleum applications are in Group D, i.e., a Class I, Group D application.

One can use an explosion-proof motor but these are seldom available above 1500 hp, although larger ones have been built. At some point, cost is prohibitive. There are several alternatives.

A very common procedure is to use a pressurized room. Clean air is drawn in to keep the air free of explosive mixtures and to carry off motor heat. This room is usually isolated from equipment containing hazardous liquid or vapor. As a general rule, minimum air requirements are about $0.3 \text{ m}^3/\text{min}/\text{m}^2$ of floor area. Some value greater than this may be necessary to account for air leakage from the enclosed area.

An alternative to air purging is to use a totally enclosed motor filled with inert gas. These are complicated, an operating nuisance and are seldom used.

Applicable standards differ in the U. S. and Europe because of different philosophies about how to approach safety. One must reconcile these differences in planning.

One must be concerned about many standards, including motor temperature rise. For example, a Class I, Group D explosion-proof motor must maintain a surface temperature below 215°C during any load condition. This and other such standards will be handled by the specialists but we must be generally familiar with the implications of these standards as they affect planning, specification and design.

Flexible couplings are used most commonly with electric motors. They come in many forms: flexible-disc, gear, spring-grid, pin and bushing and rubber-biscuit types. The choice depends on many factors. Do not misinterpret the word "flexible." Precise alignment is critical to achieve satisfactory performance.

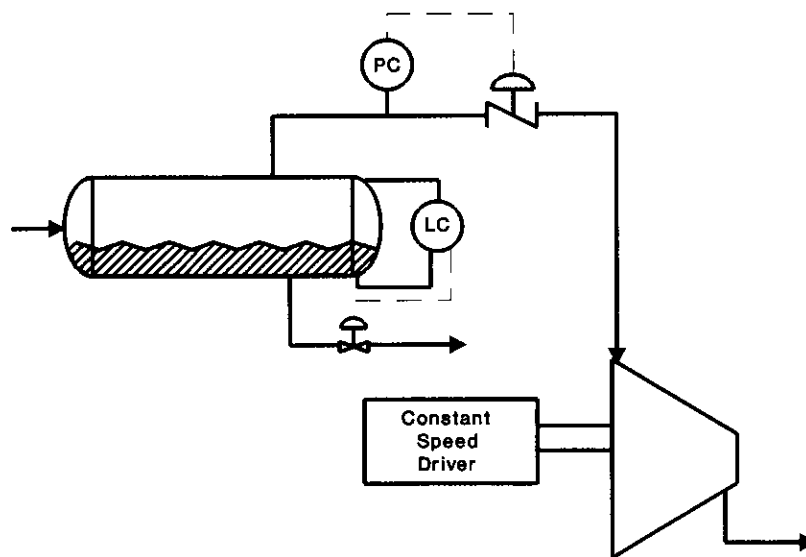
CONTROL OF COMPRESSORS

Control systems for compressors are well documented by vendors of the equipment used. Most control problems thus seem to stem primarily from errors of omission. Some vendor-supplied systems are more nearly designed to protect the machinery than to provide an efficient, low-maintenance installation. Design specifications fail to properly denote changes in gas conditions and quantity. Too often, even if the unit runs a satisfactory amount of time, it does so under inefficient conditions. The solution – a very pragmatic appraisal of the control system in light of expected process conditions, which vary for each installation. Use of a "standard" control system never yields optimum performance, except by coincidence.

Capacity Control

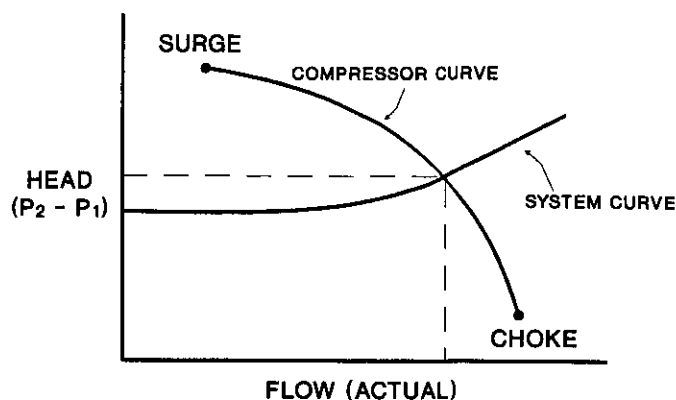
To a large extent, a compressor is self-controlling with respect to capacity. Any compressor (positive displacement or dynamic) will seek an operating condition such that its throughput matches the gas available. If the gas available is reduced, the compressor will draw down the suction pressure until its capacity again matches availability. It is only when the operating conditions fall outside the operating limits of the compressor, e.g. surge, choke, driver power, frame loads, discharge temperature, etc., that control of capacity is required.

Constant Speed Centrifugal Compressors

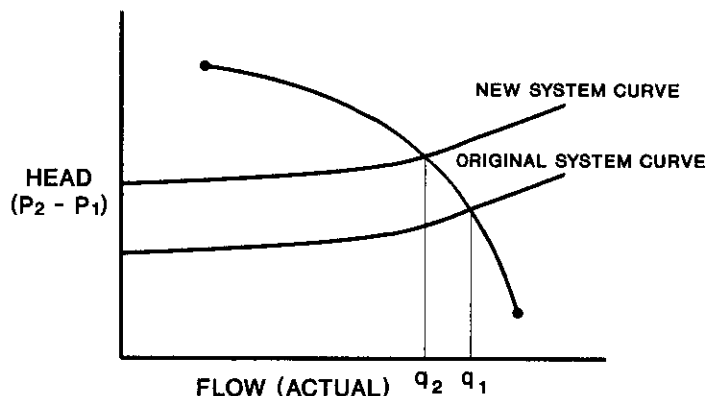


The above figure shows a typical scheme where a centrifugal compressor takes suction from a production separator with back-pressure control. This discussion is not limited to this scenario, the production separator could be any process unit requiring constant pressure operation. For a given production rate the

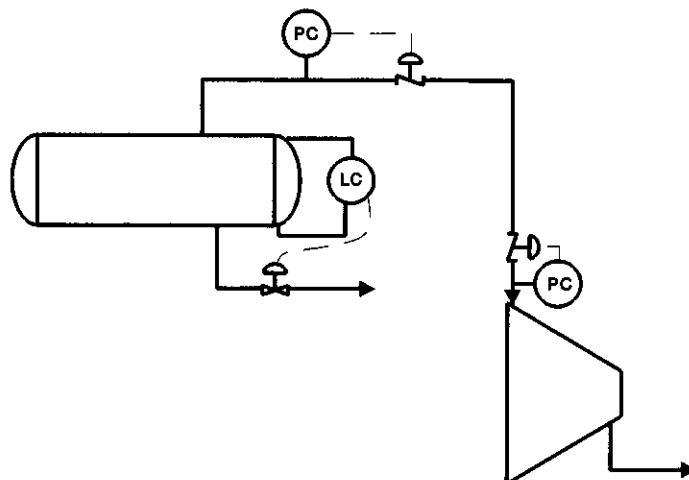
compressor will seek a suction pressure where its capacity matches availability. This is shown below using the compressor performance curve.



If the production rate is reduced, the compressor will draw down the suction pressure reducing its capacity from q_1 to q_2 as shown below. Not only has the actual volumetric throughput of the compressor decreased but the mass flowrate has declined even further since the suction pressure is now lower. Conversely if the production rate is increased, the suction pressure will rise, increasing compressor throughput.

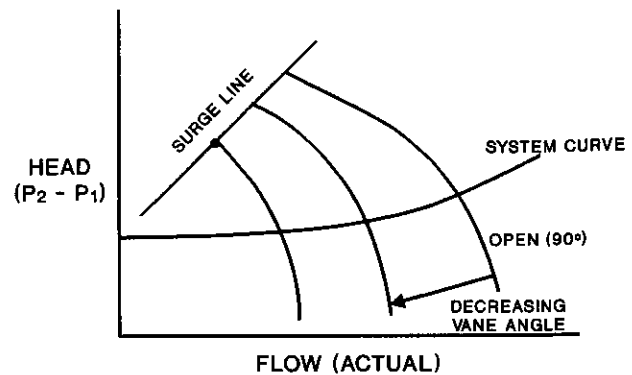


Two operating limits are imposed on this system – surge and high driver load. Surge is prevented by recycling gas from the discharge to the suction and will be discussed in greater detail later. High driver load requires a second control valve in the compressor suction to limit throughput. This valve is manipulated by a suction pressure controller with a setpoint at the throughput which coincides with the maximum driver power. Under normal operating conditions this valve is completely open.

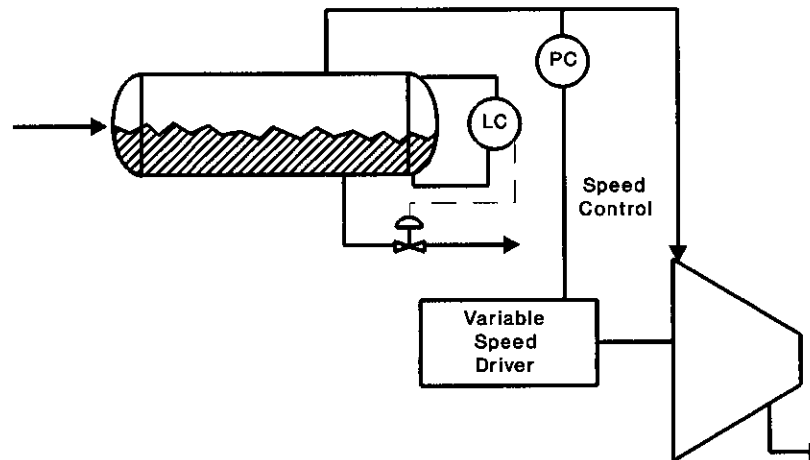


However, when the compressor throughput is limited by available driver power, this second valve controls capacity. The backpressure valve on the separator will go to the full open position and unless production is cut or gas flared the pressure on the separator will rise to the relief valve setting.

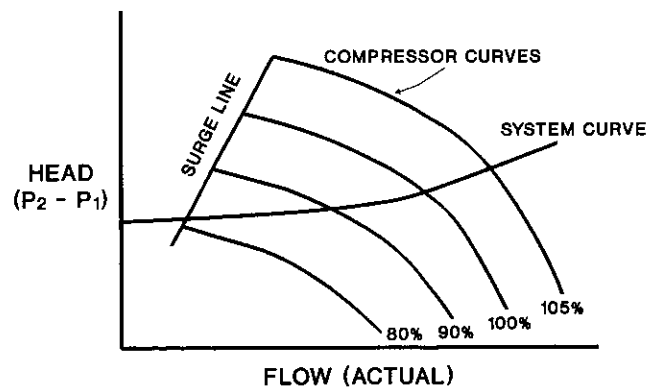
Adjustable inlet guide vanes may also be used to control compressor capacity. They are frequently used on constant speed centrifugal compressors. Adjustable inlet guide vanes essentially change the aerodynamic design of the first impeller. This is accomplished by *prewhirling* the gas entering the wheel in the direction of rotation, thereby developing less head than design. The efficiency declines as well but not as much as the head so the net result is lower driver power. These devices are more efficient than suction throttling, but are expensive and can increase maintenance expense. The figure at right shows the effect of adjustable inlet guide vanes on the shape of the compressor performance curves.



Variable Speed Centrifugal Compressors



When using a variable speed driver such as a gas turbine the compressor capacity is controlled by manipulating the driver speed. This has the added advantage of saving one control valve (back pressure valve on the separator) since the suction pressure remains constant. The performance curve for such a system is shown below.



In the drawing, the flowrate through the compressor is represented at any speed by the intersection of the system curve and compressor curve.

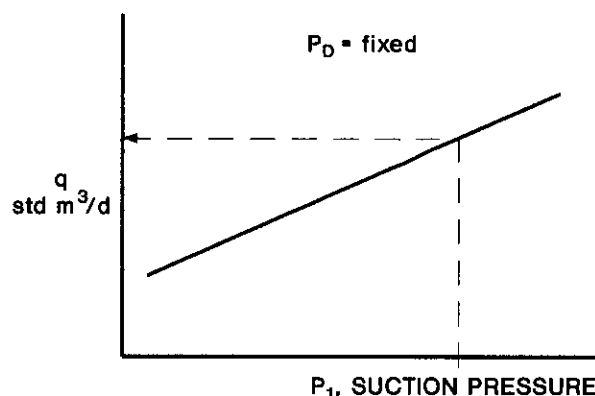
As discussed in the constant speed section, a second valve in the compressor suction may also be required to prevent driver overload.

Reciprocating Compressors

The following mechanisms may be used to control the capacity of reciprocating compressors.

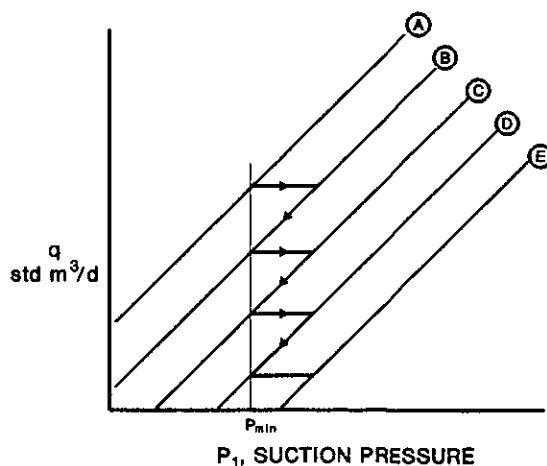
1. Suction pressure
2. Variation of clearance
3. Speed
4. Valve unloading
5. Recycle

The use of suction pressure to control capacity is virtually identical to that outlined for constant speed centrifugal compressors. A typical compressor capacity curve for a reciprocating compressor is shown to the left. As the suction pressure falls the compressor capacity (in std m³/d or ft³/d) declines due to the lower P_1 (see Equation 15.22) and the lower E_{vz} .



This compressor is self controlling as long as the operating limits of the compressor are not reached. The operating limits for a reciprocating compressor are high rod, pin and frame loads or high discharge temperature (this occurs as minimum suction pressure) and high driver load (this typically occurs at maximum suction pressure).

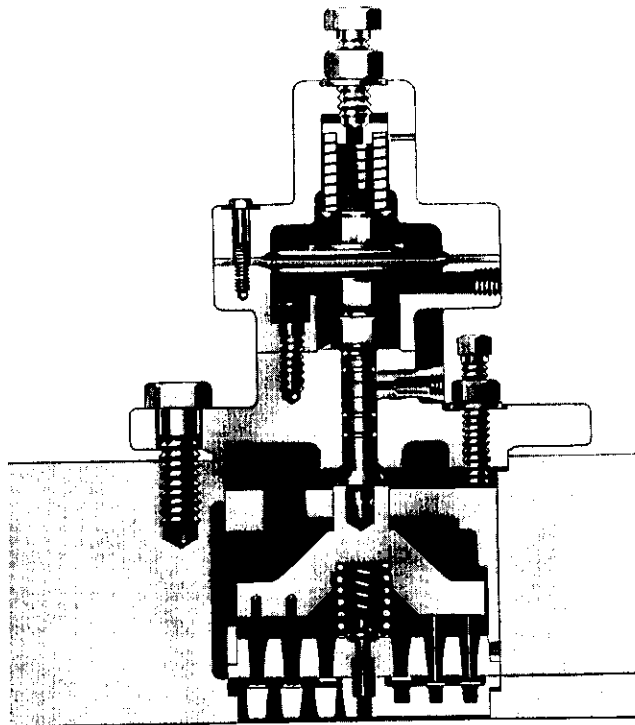
When the minimum suction pressure is reached, the compressor capacity must be reduced. This can be accomplished by increasing clearance, unloading valves and/or decreasing speed. In any case the effect is to change the compressor capacity curves as shown on the left. Curve A represents the condition where all cylinders are double-acting, all clearance pockets closed and maximum speed. Curves B-E represent a number of conditions where clearance pockets may be open, valves unloaded or speed decreased.



For some drivers, particularly electric motors and two-cycle engines, speed control is not viable. In these cases capacity control is accomplished with clearance pockets and valve unloaders.

Clearance pockets increase the clearance, decreasing volumetric efficiency, thus lowering the capacity of a cylinder. They may be fixed volume or variable volume. Fixed volume pockets are more common. They can be actuated pneumatically or manually.

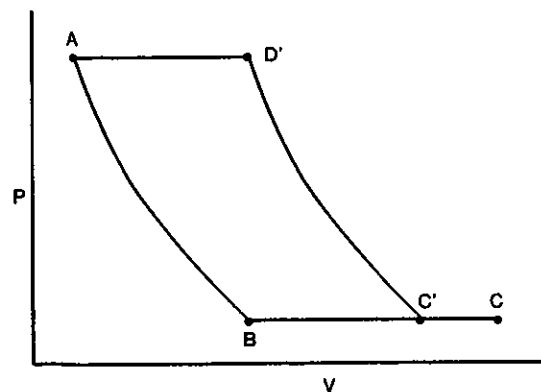
Valve unloaders decrease the capacity by holding the suction valve open through the compression cycle. As a result, no compression takes place since the gas simply flows back and forth across the suction valve. A pneumatic valve unloader is shown below.



Courtesy Cooper Industries

Because valve unloading affects the rod and frame loads, they should be used only with the manufacturer's approval.

Clearance pockets and valve unloaders are effective capacity control mechanisms, however, they provide a discrete change in capacity. This may lead to unstable control, e.g. cycling. "Stepless" unloaders are available. They hold the suction valve open through only a portion of the compression stroke. When the suction valve closes at point C' , the compression and discharge segments of the cycle occur (BC' vs. BC). The net effect is lower capacity.



More frequently, recycle is used to control compressor capacity. In recycle, gas is recirculated from the discharge to the suction side of the compressor to maintain constant suction pressure. It must be cooled to avoid increasing the suction temperature. Control is simple, straightforward and continuous, but energy inefficient.

If recycle is used it should be minimized. This can be accomplished by using clearance pockets or valve unloaders in conjunction with recycle. For example, if the recycle valve is more than 25% open, a clearance pocket will open. This reduces compressor capacity and the recycle valve must close to reduce recycle rate. Control is still continuous and simple, but more efficient.

The maximum capacity through the compressor is set by driver power. This typically occurs at maximum suction pressure. Driver overload is prevented by installing a control valve in the compressor suction to limit throughput. This is identical to that discussed under centrifugal compressors.

Anti-Surge Control

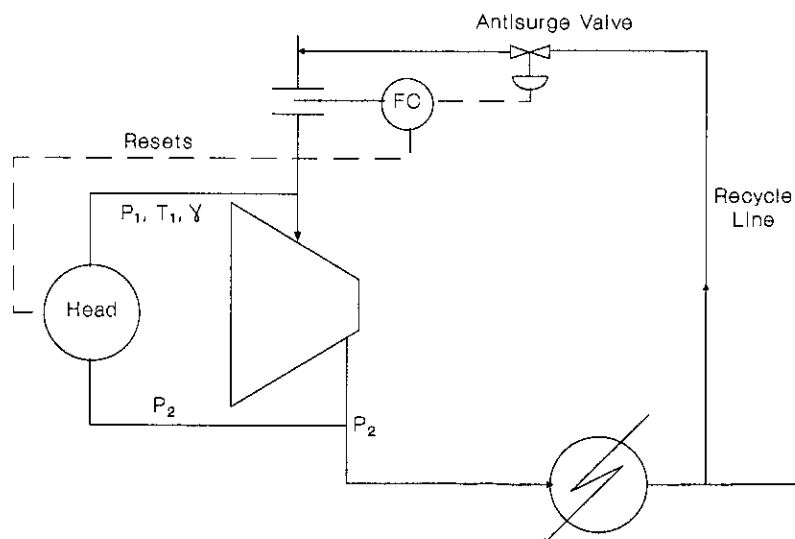
Centrifugal and axial compressors have a characteristic combination of maximum head and minimum flow beyond which they will surge. As discussed previously surge is an unstable region where flow reversal occurs inside the compressor. Surge can damage seals and bearings and in some cases can result in a catastrophic compressor failure.

The only way to prevent surge is to recycle or blow-off a portion of the flow to keep the compressor away from its surge limit. Blow-off is not feasible in natural gas compression so recycle is the most common antisurge method.

As discussed earlier, recycle is inefficient and expensive. A well designed antisurge system will protect the compressor from surge but not recycle excessive amounts of flow.

A typical surge line is shown in Figure 15.9. Its location must be determined by test prior to design of antisurge system. An algorithm is then developed to model the relationship between head and flow along the surge line. A surge control line (SCL) is then developed which mirrors the shape of the surge line but is offset to the right by a margin of safety (usually 5-10%). The closer the SCL is to the surge line, the less margin of safety and the higher the probability of surge. The further the SCL is from the surge line the greater the protection against surge but the greater the recycle resulting in less efficient operation.

The general approach to antisurge control is to measure head (or a surrogate of head, e.g. ΔP , compression ratio, etc.) and reset the setpoint on the antisurge controller as shown below:



The actual compressor head can be computed from Equation 15.5. This requires measurement of P_1 , P_2 , T_1 , and γ . Correlations for z and k are included in the software. If variations in gas composition and temperature are minimal, measurement of P_2 and P_1 is often sufficient.

Manufacturers of antisurge controllers provide their own special algorithms, and response characteristics to give safe operation with the SCL as close to the surge line as possible. Many integrate surge control with capacity control, particularly on compressors operating in series and/or parallel.

SELECTION OF COMPRESSORS AND DRIVERS

In the preceding pages, I have discussed the many variables that affect selection of compressors and drivers. Summarized below are some other basic factors for consideration.

The usual choice is centrifugal versus reciprocating. The basic unit consideration is gas turbine-centrifugal combinations or an all reciprocating unit. It is possible, but not recommended, to drive a reciprocating compressor with a gas turbine using a speed reducer. Several small units of this type have been installed.

The decision is based on process needs, capital costs and operating costs. A rational final choice will balance these considerations. It is impractical here to weigh all of the variables involved. Summarized below are some of the major considerations involved in this appraisal.

1. *Weight.* – Dynamic or kinetic devices are much lighter per power output. They also possess less vibration. Low weight eases transportation problems, particularly to remote areas. It also minimizes offshore platform costs. The weights shown below illustrate a general range for various complete compressor packages in the range of 2200-900 kW:

Aircraft turbine-centrifugal compressor, 9-10 kg/kW

Industrial gas turbine-centrifugal compressor, 13-30 kg/kW

Integral compressor, 38-52 kg/kW

2. *Initial Cost.* – Centrifugal devices usually will have a capital cost substantially less than a comparable reciprocating unit.
3. *Efficiency.* – The overall efficiency of a reciprocating unit is higher than that for a centrifugal unit – in some cases as much as 15-17% higher.
4. *Fuel Consumption.* – Heavy duty integral units offer greater fuel efficiency. The following comparison is typical:

Low-medium Speed Integral	9000-13 400 kJ/kW·hr
High Speed Reciprocating	10 000-19 000 kJ/kW·hr
Gas Turbines	10 000-20 000 kJ/kW·hr

5. *Maintenance.* – Maintenance problems tend to be more routine but often more frequent on reciprocating machines. This is primarily an experience factor. By careful attention, maintenance may be satisfactory for a centrifugal unit.

The above brief summary indicates that the type of unit should be tailored to the need. The trend toward centrifugal equipment does not render obsolete the reciprocating needs. If space and weight are not critical concerns, a reciprocating unit may show positive advantage at high compression ratios. Above a compression ratio of 1.4-1.5, the reciprocating unit may show positive advantage – particularly on a long-life project.

Below about 1000 kW, a high-speed reciprocating unit might be a good choice. Above 1000 kW, a slower-speed integral unit or centrifugal would likely be the choice. Considerations like salvage value, maintenance, fuel costs, project life, and the like, will govern.

With gas turbines, the choice is between the light "aircraft type" and industrial turbines. The former are cheaper, lighter, and are more accessible for maintenance. Historically, their reliability has not been as great as for industrial gas turbines, but recent performance records have been satisfactory. The latter, being more rugged, may have longer periods of operation between overhauls. Aircraft type, using gas as fuel, can be expected to have an overhaul at 20 000 hour intervals. This is a good planning number.

As pointed out in Table 15.1, there are many possible combinations of compressors and drivers. The final choice may depend on very practical matters like fuel quality and type.

MAINTENANCE AND SURVEILLANCE

Compressor Maintenance and Surveillance

Many operators are finding that routine surveillance and monitoring – continuously or at regular periods – pay dividends, particularly on large installations. Electronic analyzers have proven useful for this purpose. They provide power on all cylinders, engine rpm, volumetric efficiency, vibration at key points, ignition information and pressure-volume information. All of this is useful for preventive maintenance.

Torque may be measure, both directly or by an electro-mechanical system involving no direct contact.

It is desirable to minimize tearing down, inspecting and cleaning compressor components. This is not only expensive but injects the possibility of improper reassembly, getting dirt into bearings, etc., etc. The trade literature abounds with measurements and techniques for this purpose that do not require extensive equipment.

Vibration and Pulsation

Unsteady flow of gas is an integral part of a reciprocating device as the pistons move back and forth. Pulsation flow causes decreased operational efficiency. Capacity variations of 5% are common.

Any excess pulsation is usually a result of acoustic resonance or standing-wave buildup in the piping. Vibration will result when the frequency of a pulse coincides with the resonant frequency of the piping. Continuing study of this area is desirable. Available analytical methods are not adequate.

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- 15.3 Balje, O. E., *J. Eng. Power* (Jan. 1962).

16

REFRIGERATION SYSTEMS

A refrigeration system lowers the temperature of the fluid being cooled below that possible when using air or water at ambient conditions. A typical building air conditioner cools air to a temperature of 10-15°C. At the other end of the scale is the liquefaction of helium at -268°C. The temperature produced depends on the goal of the exercise. If the goal is to produce marketable liquids, basic economics controls the temperature specified. If it is to meet a hydrocarbon dew point, that specification sets forth the required temperature.

Several basic processes will be discussed herein.

1. Absorption refrigeration
2. Compression refrigeration
3. Expansion across a turbine
4. Expansion across a valve

AMMONIA ABSORPTION SYSTEM

Figure 16.1 shows a flowsheet for a refrigeration system utilizing using two concentrations of ammonia-water solutions. The basic driving force is the heat input to the generator. Ammonia vapor is stripped from the water solutions in the rectifier or stripper. This ammonia vapor is condensed and passes through a receiver, a heat exchanger (optional) and across an expansion valve into the evaporator or chiller. Here it vaporizes while cooling the fluid to be chilled.

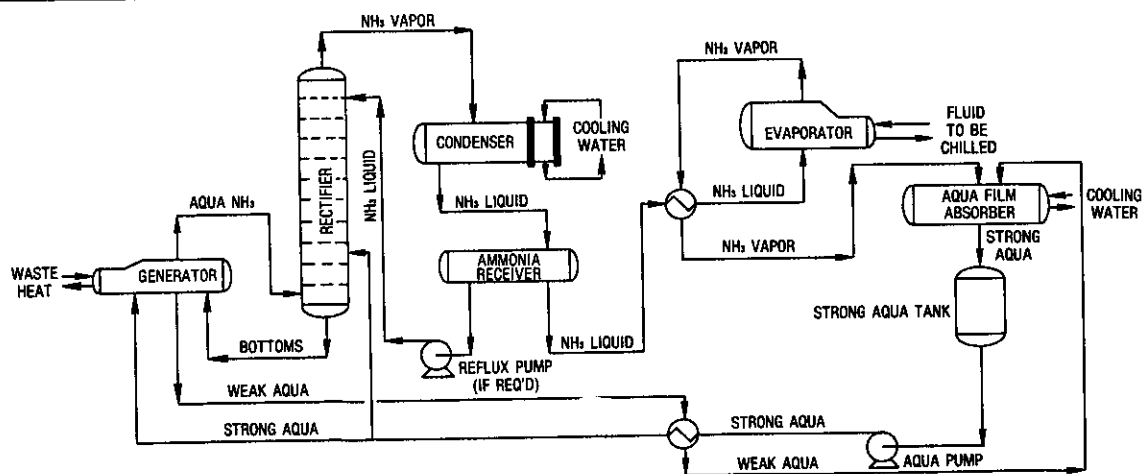


Figure 16.1 Flow Sheet of an Ammonia Absorption System

The ammonia vapor from the evaporator is absorbed in a weak ammonia-water solution. The result is a strong solution. The ammonia is removed from this strong solution in the generator to begin its cycle all over again. Circulation of the strong solution is usually performed with vertical multi-stage pumps.

One can write selected energy balances around this system to determine loading at various points. The starting point is the evaporator. First, determine the total H required to chill the fluid to its desired temperature. This is the heat load in the evaporator.

The heat absorbed by the ammonia per unit mass is governed by evaporator pressure or temperature. The ΔH is the enthalpy of a saturated ammonia vapor at evaporator conditions minus the enthalpy of the entering ammonia, which should be a saturated or subcooled liquid.

A series of such balances, and an overall balance, enable one to determine sizes and energy loads of each component part. Table 16.1 summarizes the typical loadings for the ammonia system in Figure 16.1.^(16.1)

TABLE 16.1
Utilities Requirements for Ammonia Absorption System

Evap. temp. °F	50	40	30	20	10	0	-10	-20	-30	-40	-50
Steam pressure, psia	14.1	19.8	24.9	30.9	41.8	53.2	67.0	83.2	103.1	134.6	173.3
Single Stage											
Steam sat. temp., °F (or waste heat exit temp.)	210	225	240	255	270	285	300	315	330	350	370
Generator heat required Btu/min/TR	300	325	347	373	400	430	466	511	571	645	754
Steam rate, lb./hr./TR	18.9	20.2	21.8	23.6	25.8	28.1	30.9	34.1	39.1	44.6	53.2
Water rate thru cond. & absorber, gpm/TR 85°F on 105°F off	3.6	3.7	3.8	4.0	4.3	4.5	5.0	5.5	6.1	7.1	8.8
Two Stage											
Generator's steam sat. temp. °F, exit temp.	175	180	190	195	205	210	220	230	240	250	265
Steam pressure, psia	6.7	7.5	9.3	10.4	12.3	14.1	17.1	20.7	24.9	29.8	38.1
Generator heat required Btu/min/TR	550	577	605	637	670	711	753	799	850	905	970
Steam rate, lb./hr./TR	33.2	34.9	37.0	39.1	41.3	43.9	46.7	49.9	53.6	57.5	62.3
Water required thru cond. & absorber, gpm/TR 87°F on 105°F off	4.0	4.2	4.3	4.5	4.9	5.3	5.8	6.4	7.2	8.3	10.2

Evap. temp. °F	10	5	0	-5	-10	-15	-20	-25	-30	-35	-40	-45
Steam pressure, kPa	97.2	132.3	164.7	200.5	258.1	327.5	404.8	495.8	601.0	734.1	928.0	1166.3
Single Stage												
Steam sat. temp., °C (or waste heat exit temp.)	99	106	114	121	129	136	144	151	158.9	167	177	187
Generator heat required kJ/m/TR	317	340	361	385	411	438	469	506	552	611	680.5	783
Steam rate, kg/h/TR	8.6	9.1	9.8	10.5	11.3	12.2	13.2	14.5	15.9	18.0	20.2	23.7
Water rate thru cond. & absorber, m ³ /h/TR 29°C on, 41°C off	0.82	0.84	0.86	0.89	0.95	1.00	1.07	1.17	1.28	1.41	1.61	1.99
Two Stage												
Generator's steam sat. temp. °C, exit temp.	75	80	85	90	95	100	105	110	115	120	125	130
Steam pressure, kPa	37.6	47.4	57.9	67.9	81.6	101.3	120.6	142.7	168.6	198.7	232.4	266.8
Generator heat required kJ/m/TR	536	586	624	665	700	759	800	843	891	943	987	1028
Steam rate, kg/h/TR	14.0	15.3	16.3	17.5	18.5	20.2	21.4	22.6	24.1	25.7	27.1	28.5
Water required thru cond. & absorber, m ³ /h/TR 31°C on, 41°C off	0.85	0.92	0.97	1.01	1.09	1.22	1.33	1.45	1.62	1.84	2.09	2.35

The heat for the generator may be obtained from any one of four sources: (1) low pressure steam, (2) fired heater, (3) a hot process stream and (4) waste heat. Since heat loads are large, the availability of (3) or (4) increases the economic attractiveness of this system.

Water cooling is shown for the condenser and absorber but air cooling may be used. As with all refrigeration systems, the higher the cooling temperatures the greater will be the net energy load.

Evaporator temperatures down to about -50°C [-58°F] are practical. Units have been designed with capacities to about 35 MW [10 000 tons of refrigeration (TR)]. The unit is simple and has few moving parts to maintain. Ammonia solutions are not difficult to handle metallurgically.

A major psychological problem is ammonia smell. In a confined space this can be a nuisance. However, the pungent odor is a safety item that immediately confirms leaks.

Ammonia systems are generally competitive with compression systems in initial cost. Operating cost comparisons are dependent on the source of heat and the cost of cooling. If the economics are competitive, they are a viable alternative to compression systems for producing refrigeration.

COMPRESSION REFRIGERATION

Figure 16.2 shows the simplest compression refrigeration system. Saturated liquid at Point A expands across a valve (isenthalpically). On expansion some vaporization occurs. The mixture of refrigerant vapor and liquid enters the chiller at $3\text{--}6^{\circ}\text{C}$ [$5\text{--}10^{\circ}\text{F}$] lower than temperature to which the process stream is to be cooled. The liquid vaporizes. Leaving at Point C is a saturated vapor at the P and T of the chiller. This vapor is compressed and then enters the condenser as a superheated vapor. The condenser is cooled by water or air. The liquid then enters the accumulator and returns to the expansion valve.

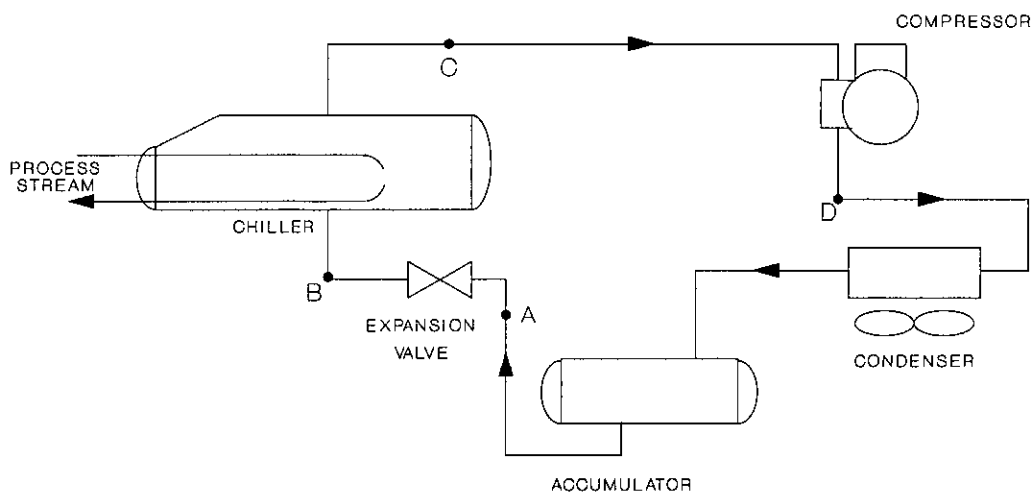


Figure 16.2 Flow Sheet of a Simple Refrigeration System

The refrigerant must leave the condenser as a saturated liquid or slightly subcooled. Nothing happens in the accumulator. It merely serves as a reservoir for refrigerant as levels vary in the chiller(s) and condenser.

Figure 16.3 is a more complex cycle composed of the same type of equipment. Assuming two stages of compression, a second expansion valve and a separator are added. This system will require less power per unit of heat load in the chiller. The reason? Part of the circulation rate is only compressed through one stage and the refrigerant entering the chiller contains less vapor. The vapor passing through the chiller does virtually no cooling, even though it contributes to compression cost.

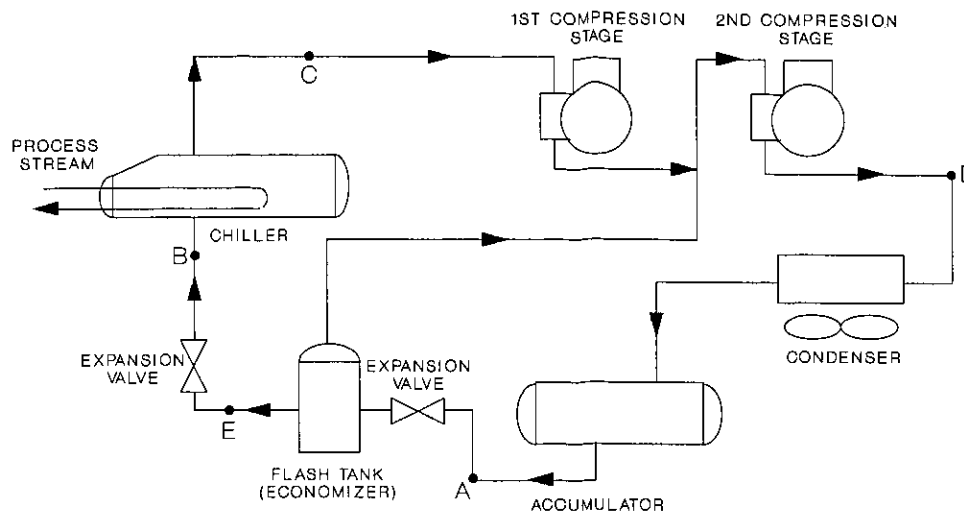


Figure 16.3 Flow Sheet of a Refrigeration System with an Economizer

The separator could be called an *economizer*, a word that applies to any device – usually a heat exchanger – that reduces net utility consumption. A separator-choke combination can be provided between each stage of separation, regardless of the number of stages.

One also can add a heat exchanger in the suction line to the compressor to exchange heat between the vapor from the chiller and the liquid to the choke preceding the chiller. This exchanger also could be called an economizer.

Calculation of a Simple System

There are several discrete steps in the sizing of the system shown in Figure 16.2. These are summarized below:

1. Determination of refrigerant circulation rate -

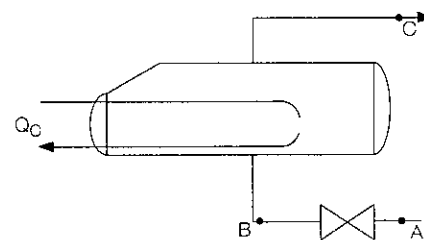
The balance at right is around the chiller and expansion valve. At Point A the refrigerant is a saturated liquid (or very close to it). At Point C it is a saturated vapor. Q_C is the heat load determined by specifications on the stream being cooled.

If one writes an energy balance around the system, $Q_C + m_A h_A = m_C h_C$. But $m_A = m_C = m$, so

$$m = Q_C / (h_C - h_A) \quad (16.1)$$

Where:

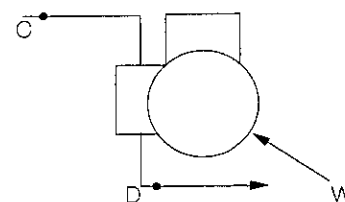
Q_C = chiller heat load
 h_C = saturated vapor enthalpy
 h_A = saturated liquid enthalpy
 m = circulation rate



Metric	English
kJ/kg	Btu/lb
kJ/kg	Btu/lb
kJ/kg	Btu/lb
kg/s	lb/hr

2. Determination of Compressor Power -

This is done by any appropriate method outlined in Chapter 15. One will calculate theoretical (isentropic) work and use an efficiency to find actual work. The circulation rate from Step (1) is used.

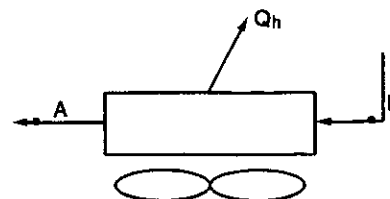


3. Determination of Condenser Heat Load (Q_h) -

There are two ways to do this. Knowing Q_c and W , you can write an overall balance as shown in Figure 16.2 to find Q_h .

If you are performing the calculation manually and wish an independent check of the previous work, write the balance shown at right.

$$Q_h = m(h_A - h_D) \quad (16.2)$$

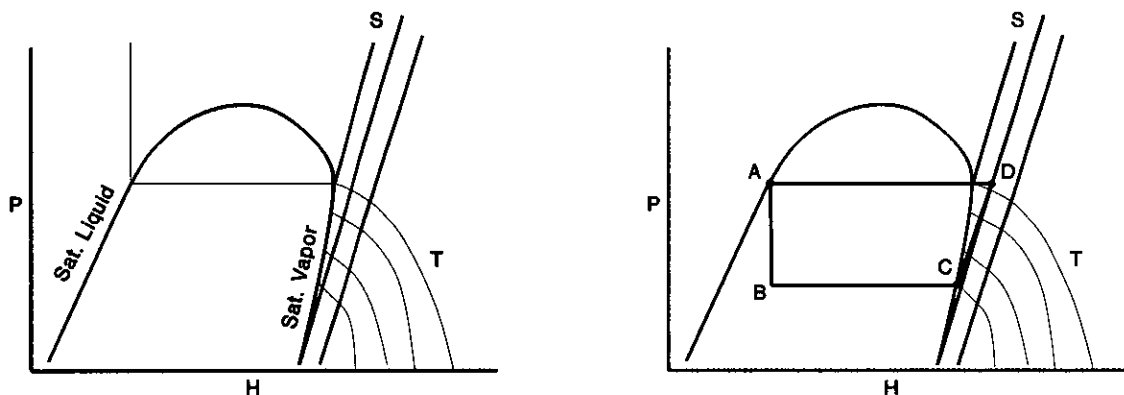


The Q_h found from Equation 16.2 will be negative. This merely signifies that heat is leaving the system. The value found from the overall balance will not check the condenser balance exactly because only part of the compressor inefficiency shows up in the refrigeration system. For practical purposes, the difference is trivial.

Determination of the Enthalpies

The calculation requires one to find the enthalpy per unit mass at Points A, B and C. This can be done from a computer routine. Convenient tables and figures are available for all of the common commercial refrigerants. Appendix B at the end of this volume contains data on substances used as refrigerants. Appendix 16A at the end of this chapter contains pressure-enthalpy (P-H) figures for propane and Refrigerant 22 (a Freon) as well as vapor pressure and physical property information on all common refrigerants.

The P-H diagram is very convenient for solving the energy balance for a simple system.



The left-hand figure is a representation of the P-H diagrams in Appendix 16A. The refrigerant is all liquid to the left of the saturated liquid curve; it is two-phase inside the saturation curve and all vapor to the right of the saturated vapor curve. The lines of constant temperature are horizontal between the saturated vapor and liquid curves and then rise almost vertically in the liquid section.

The calculation process starts by choosing the temperature of Point A. Will water, air or some other stream be used for condensation of the refrigerant? What temperature can we realistically achieve in the condenser? That is Point A. It is on the liquid saturation curve, since it leaves the condenser as a liquid.

What is the temperature at Points B and C? Normally, it will be 3-6°C less than the minimum desired temperature for the fluid being cooled. This approach fixes the location of Point C. It is on the saturated vapor curve, since it is in equilibrium with the liquid in the chiller (evaporator).

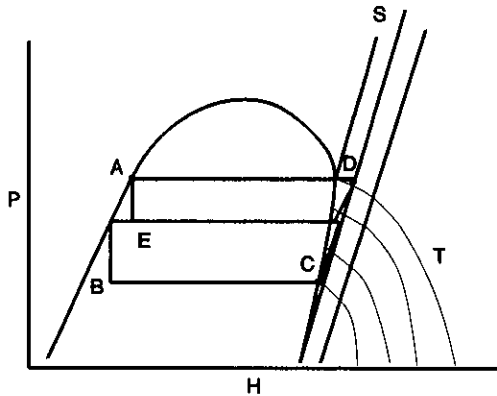
The expansion across the choke from Point A is an *isenthalpic* process; a vertical line on a P-H diagram. Draw a vertical line from A to B, the pressure of Point C, and then go horizontally to C. One can read the Δh required for Equation 16.1.

The theoretical compression is *isentropic*. Starting at Point C, parallel the constant entropy lines until you intersect the pressure line of Point A. This is theoretical Point D.

$$W_{\text{theor}} = (m) (h_D - h_C)$$

Equation 16.2 for condenser heat load is found from the Δh between Points D and A.

For a commercially pure refrigerant, use of a P-H diagram or a corresponding table is as reliable as any method.



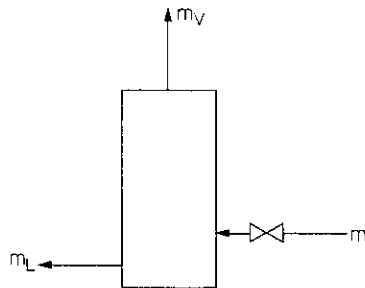
The *economizer system* shown in Figure 16.3 can be calculated in the same manner.

The P-H diagram is shown at left. Point E is at a pressure that allows the compression ratio to be the same in each stage of compression. Expansion from A to E is isenthalpic. The vapor formed goes to the second stage of compression as a saturated vapor. The saturated liquid leaving the separator is expanded isenthalpically to pressure B. Notice that the Δh available from B to C for this system is larger than for the simple system.

Calculation of Economizer Systems

Determination of circulation rate, work and Q_h for the system in Figure 16.3 follows the same pattern as for the simplest system.

The first step is the same in principle. Now, however, the enthalpy of the refrigerant entering the choke is determined by compressor interstage pressure and not by condenser pressure. With this change in h_A , Equation 16.1 may be solved for " m_L ," the circulation rate to the chiller (and the amount of gas to be compressed through Stage 1 of the compressor).



What is the circulation rate through the condenser? It is " m ," where $m = m_L + m_V$. If you write the balance shown at left,

$$m h = m_L h_L + m_V h_V$$

Since $m = m_L + m_V$, one can take $m = 1.0$ and then define x and y as the relative amounts of liquid and vapor, respectively, from the separator. Then,

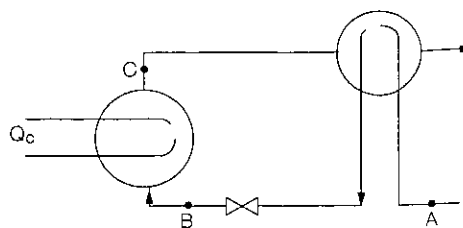
$$h = x h_L + (1 - x) h_V \quad \text{or} \quad x = \frac{h_V - h}{h_V - h_L} \quad (16.3)$$

The three enthalpies can be obtained, which enables one to solve for " x ," and then " y ." Knowing these relative quantities and m_L , m and m_V may be calculated.

In the compressor work calculation, one finds the work in the first stage (for flow rate " m_L ") and adds it to second stage work (for flow rate " m ") to find total work. The resultant temperature of the gases after they mix in the tee between stages may be found by a balance around that tee. In most cases, the temperature effect here is negligible.

The condenser heat load is found as before.

If one places a heat exchanger to exchange heat as shown, an extra balance is needed. The cold vapor will subcool the saturated liquid from A and there will be less vaporization upon expansion across the valve. However, the gas will go to the compressor at a higher temperature and lower pressure. Is the investment worthwhile? Only a calculation will tell. One might add the heat exchanger shown to the system in Figure 16.3.



Calculation of Chiller Load (Q_C)

The refrigeration load must be calculated from the specifications on the system in which the chiller is placed. Figure 16.4 shows a simple system using glycol injection to inhibit hydrate formation.

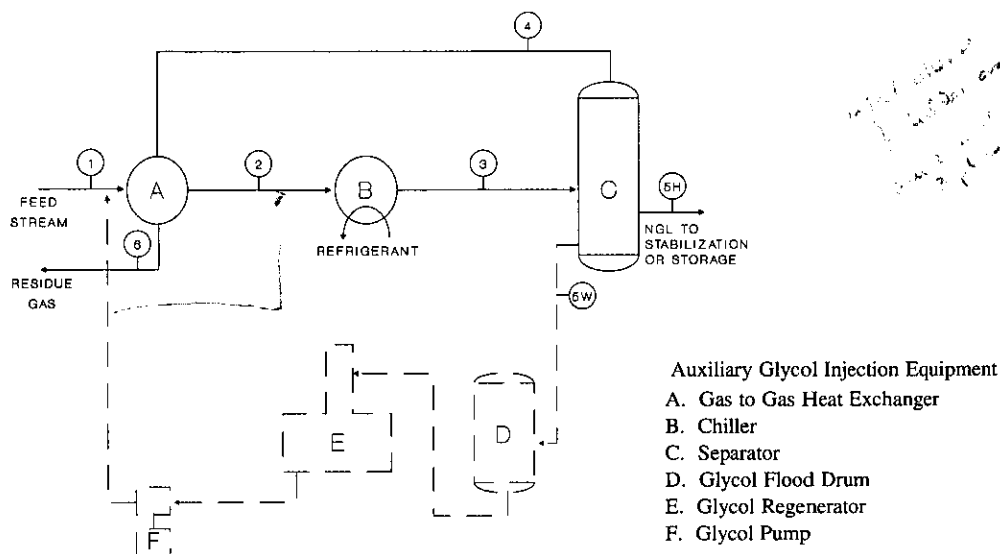


Figure 16.4 Schematic View of a Compression Refrigeration System Using Glycol Injection

The first step in the design is to fix the temperature (T_3) in the low temperature separator (LTS). The pressure in the LTS must be high enough above the specified sales pressure to allow for pressure drop in the HEX and lines.

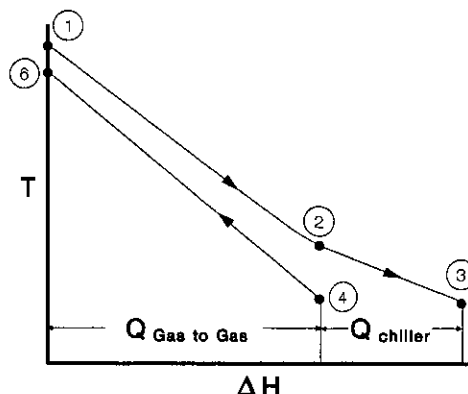
The minimum temperature coming to the HEX is fixed by the economics of precooling the feed stream. The maximum sales gas temperature is usually fixed by contract and is seldom allowed to exceed 50°C [122°F]. Consequently, the heat load between P_1 , T_1 and P_3 , T_3 is fixed by these considerations. The problem revolves around the distribution of this load between the HEX and the refrigerated chiller. For this calculation, it is convenient to assume a 34 kPa [5 psi] drop in each heat exchanger. The following general procedure is suggested if the HEX and chiller are to be sized as part of the exercise.

1. Determine the amount and composition of vapor and liquid at T_1 , P_1 and T_3 , P_3 . (Don't forget the water and glycol.)
2. Calculate the total ΔH between points 1 and 3 (H_1 and H_3).
3. Calculate the cooling capacity for the sales gas in the gas-to-gas exchanger ($H_6 - H_4$), T_6 should be fixed at ($T_1 - 5^\circ\text{C}$) or the contractual maximum temperature, whichever is lower.
4. Calculate the chiller duty, Q_C , by difference

$$Q_C = (H_1 - H_3) - (H_4 - H_6)$$

5. Perform a series of flashes on the feed stream at temperatures and pressures between points 1 and 3. Plot ΔH versus T .

Using T_4 , H_4 and T_6 , H_6 plot ΔH versus T for the residue gas.



This graph is used to determine the temperatures necessary for sizing heat exchangers. It may be generated by hand using the procedure outline previously or developed by computer simulation.

The cold liquid from the LTS is also available for cooling service since usually it must be heated before entering the fractionation system. Although not shown in Figure 14.7, it may be used for cooling the feed or for any other cooling function within the system. If this liquid is not heated, a cold-feed stabilizer of the type discussed in Chapter 17 might be specified.

Specification of Liquid Separation Temperature

The desired temperature of the gas-liquid stream leaving the chiller must be determined by the strategy governing the system. If hydrocarbon dew point control is primary and liquid recovery secondary, this temperature should be about 3-5°C [5-9°F] below the temperature required to achieve the specified dewpoint. This required temperature may be significantly colder than the specified dewpoint if the gas is processed at high pressure.

If liquid recovery is the primary function of the unit, what products are salable? The basic strategy is to condense the least amount of nonsalable components (usually methane or ethane) compatible with the economics. Anything condensed, like methane, that must be revaporized and maybe recompressed, adds to the operating costs without contributing to liquid revenue.

For a given set of specifications, one should investigate a series of LTS separator pressures and temperatures. The pressure for maximum liquid recovery is between 3.0-4.0 MPa [435-580 psia], if propanes plus are the salable product. As the pressure increases, condensation of methane increases. However, 3.0-4.0 MPa [435-580 psia] may not be the optimum pressure economically because of system logistics. The optimum pressure must minimize total system cost, not merely that of the refrigeration system alone. As a general rule, separation is carried out at necessary sales gas pressure to eliminate recompression.

For the usual pressures chosen, what is a reasonable temperature? As noted before, this depends on the products desired. If the liquid product is to be stable at atmospheric pressure and sold as "crude oil," a common optimum separation temperature is 0-5°C [32-40°F]. When propane is the lightest salable liquid, the temperature may be -40 to -18°C [-40°F to 0°F]. This temperature depends on the recovery desired and whether absorption or adsorption is combined with refrigeration. Below about -40°C [-40°F] you get into the cryogenic range for ethane recovery.

Choice of temperature (and pressure) is a critical specification. Do not choose arbitrarily? Calculate the economics for a series of conditions and choose the optimum one.

Choice of Refrigerant

The ideal refrigerant is nontoxic, noncorrosive, has PVT and physical properties compatible with the system needs, and has a high latent heat of vaporization. Any material could be used as a refrigerant. The practical choice reduces to one which has desirable physical properties and will vaporize and condense at reasonable pressures, at the temperature levels desired. The usual choice is propane, ammonia, R-12 or R-22 at chiller temperatures above about -40°C . At cryogenic conditions, ethylene and methane might be used. In general, the lower practical limit of any refrigerant is its atmospheric pressure boiling point. It is desirable to carry some positive pressure on the chiller to obtain better efficiency in the compressor, reduce equipment size and avoid air induction into the system.

Table 16.2 furnishes an excellent comparison of the common refrigerants. Consider, for example, a unit where the evaporator (chiller) operates at -18°C [0°F] and the condenser at 35°C [95°F]. Ammonia requires the lowest mass circulation rate, but propylene and propane have slightly lower horsepower requirements. This table also illustrates the effect of condensing temperatures on horsepower and circulation needs. Raising the condensing temperature 17°C [31°F] increases horsepower requirements about 60% for propane and 43% for ammonia. There are two morals – (1) use the lowest temperature possible for condensing to minimize cost; and (2) if higher condensing temperatures are required, ammonia might be preferable to propane. Ammonia is seldom chosen because of emotional reactions to its odor. However, it is easy to handle in ordinary steel equipment containing no copper and brass and is really less dangerous than propane because of its pungent odor. No dangerous accumulation can build up unnoticed.

Propane is by far the most popular refrigerant in the gas processing applications. It is readily available (often manufactured on-site), inexpensive and has a "good" vapor pressure curve. It is flammable but this is not a significant problem if proper consideration is given to the design and operation of the facility.

Freons are widely used as commercial refrigerants. They are non-toxic and nonflammable. CFC (Chlorofluorocarbon) refrigerants like R-11 and R-12 are being phased out due to environmental problems. HCFC refrigerants (Chlorofluorocarbons containing at least one hydrogen) are currently considered environmentally acceptable. R-22 is the most popular although other HCFC's are being developed as replacements for R-11 and R-12. Freons are expensive and system losses can represent a significant operating cost. They are also difficult to ship to remote locations in large quantities. Certain freons will form hydrates so it is necessary to keep the system dry.

Compressor choice is linked to refrigerant choice as well as other considerations. Where weight and size are particularly important, a centrifugal or a screw compressor may be used. A reciprocating compressor is an excellent alternative for accessible land locations. The choice will depend on total power requirements.

Regardless of the choice, the controls must accommodate frequent, and sudden, load changes. As gas stream flow rate and/or composition changes, so will refrigerant circulation rate. Thus, some form of speed control is often desirable. In many cases, electric motors make an ideal driver when compressor specifications are compatible with motor characteristics.

Effect of Temperature on Cost

Figure 16.5 shows the approximate relative effect of temperature on compression refrigeration cost. It emphasizes the previous statement that one should use the highest temperature compatible with the goals of the installation. I keep emphasizing this because too many persons buy a "standard" unit without much thought about the temperature level needed. This may be a waste of money.

The inset to Figure 16.5 shows the refrigerant often used at various temperature levels. The temperature levels are approximate. Actually, the refrigerant used at a given level depends on economics, which will vary in different circumstances.

for Refrigeration
= 11374 KJ/hr
= 12,000 Btu/hr

TABLE 16.2
Comparison of Common Refrigerants

		Evaporator, Temp. °C														
		-62	-57	-51	-46	-40	-34	-29	-23	-18	-12	-6	-1	4	10	15
Evaporator pressure in kPa	Ammonia			38.3	52.9	71.6	95.8	126	163	210	265	331	411	503	615	741
	Propylene	49.6	65.5	86.1	111	143	179	221	269	331	400	482	568	662	779	903
	Propane	38.3	51.3	67.4	86.8	112	141	176	216	163	320	386	464	551	649	758
	Freon 12	19.8	27.4	36.9	49.1	64.1	82.7	105	132	164	202	246	297	356	423	499
Condensed Liquid Temperature 35°C; Condenser Pressure in kPa: Ammonia 1357; Propylene 1461; Propane 1220; Freon 12 848																
kg refrigerant per minute per ton of refrigeration	Ammonia			0.206	0.204	0.202	0.200	0.199	0.197	0.196	0.195	0.193	0.192	0.191	0.191	0.190
	Propylene	0.939	0.913	0.889	0.871	0.848	0.830	0.812	0.798	0.780	0.767	0.753	0.739	0.726	0.712	0.699
	Propane	0.990	0.958	0.925	0.898	0.875	0.853	0.830	0.807	0.789	0.776	0.756	0.739	0.721	0.708	0.694
	Freon 12	2.35	2.28	2.22	2.16	2.11	2.05	2.00	1.96	1.91	1.87	1.84	1.79	1.76	1.73	1.70
Brake horsepower per ton of refrigeration	Ammonia			4.31	3.74	3.23	2.80	2.741	2.08	1.78	1.50	1.26	1.03	0.835	0.648	0.483
	Propylene	5.00	4.47	3.96	3.51	3.10	2.69	2.35	2.06	1.74	1.46	1.20	1.00	0.830	0.647	0.485
	Propane	4.98	4.39	3.87	3.473	3.03	2.67	2.32	2.03	1.75	1.49	1.247	1.01	0.800	0.622	0.458
	Freon 12	5.70	4.98	4.33	3.79	3.31	2.86	2.47	2.14	1.83	1.55	1.30	1.05	0.848	0.668	0.490
Kilowatt per to of refrigeration	Ammonia			3.22	2.79	2.41	2.09	1.80	1.55	1.33	1.12	0.940	0.768	0.623	0.483	0.360
	Propylene	3.73	3.33	2.95	2.62	2.31	2.01	1.75	1.54	1.30	1.09	0.895	0.746	0.619	0.483	0.362
	Propane	3.72	3.27	2.89	2.56	2.26	1.99	1.73	1.51	1.31	1.11	0.925	0.753	0.597	0.464	0.342
	Freon 12	4.25	3.72	3.23	2.83	2.47	2.13	1.84	1.60	1.37	1.16	0.970	0.791	0.633	0.498	0.366
Condensed Liquid Temperature 52°C; Condenser Pressure in kPa: Ammonia 2088; Propylene 2164; Propane 1792; Freon 12 1268																
kg refrigerant per minute per ton of refrigeration	Ammonia			0.223	0.221	0.219	0.217	0.215	0.213	0.211	0.210	0.209	0.207	0.206	0.205	0.204
	Propylene	1.21	1.17	1.13	1.10	1.07	1.03	1.01	0.980	0.957	0.934	0.912	0.894	0.875	0.857	0.844
	Propane	1.30	1.24	1.19	1.15	1.11	1.07	1.04	1.01	0.980	0.953	0.925	0.903	0.880	0.857	0.835
	Freon 12	2.91	2.81	2.71	2.63	2.54	2.47	2.40	2.33	2.27	2.21	2.15	2.10	2.05	2.01	1.96
Brake horsepower per ton of refrigeration	Ammonia			5.68	4.96	4.38	3.81	3.33	2.92	2.54	2.19	1.90	1.63	1.38	1.16	0.952
	Propylene	7.49	6.72	5.96	5.32	4.71	4.14	3.66	3.23	2.79	2.41	2.03	1.78	1.55	1.31	1.10
	Propane	7.47	6.60	5.85	5.18	4.60	4.06	3.59	3.18	2.81	2.43	2.07	1.77	1.50	1.25	1.03
	Freon 12	8.09	7.11	6.25	5.46	4.78	4.18	3.67	3.20	2.78	2.41	2.07	1.77	1.49	1.24	1.02
Kilowatt per ton of refrigeration	Ammonia			4.24	3.70	3.27	2.84	2.48	2.18	1.89	1.63	1.42	1.22	1.03	0.865	0.710
	Propylene	5.59	5.01	4.45	3.97	3.50	3.09	2.73	2.41	2.08	1.80	1.51	1.33	1.16	0.977	0.821
	Propane	5.57	4.92	4.36	3.86	3.43	3.03	2.68	2.37	2.10	1.81	1.54	1.32	1.12	0.933	0.768
	Freon 12	6.04	5.30	4.66	4.07	3.57	3.12	2.74	2.39	2.07	1.80	1.54	1.32	1.11	0.925	0.761

TABLE 16.2(a)
Comparison of Common Refrigerants

		Evaporator Temp. °F														
		-80	-70	-60	-50	-40	-30	-20	-10	0	10	20	30	40	50	60
Evaporator pressure in psia	Ammonia			5.55	7.67	10.4	13.9	18.3	23.7	30.4	38.5	48.0	59.7	73.0	89.2	107.5
	Propylene	7.20	9.50	12.5	16.1	20.7	26.0	32.1	39.0	48.0	58.0	70.0	82.5	96.0	113	131
	Propane	5.55	7.45	9.78	12.6	16.2	20.5	25.5	31.3	38.1	46.4	56.0	67.3	80.0	94.1	110
	Freon 12	2.88	3.97	5.36	7.12	9.3	12.0	15.3	19.2	23.8	29.3	35.7	43.1	51.7	61.4	72.4
Condensed Liquid Temperature 95°F; Condenser Pressure in PSIA: Ammonia 197; Propylene 212; Propane 177; Freon 12 123																
Pounds of refrigerant per minute per ton of refrigeration	Ammonia			.454	.450	.446	.442	.438	.435	.432	.429	.426	.424	.422	.420	.418
	Propylene	2.07	2.01	1.96	1.92	1.87	1.83	1.79	1.76	1.72	1.69	1.66	1.63	1.60	1.57	1.54
	Propane	2.18	2.11	2.04	1.98	1.93	1.88	1.83	1.78	1.74	1.71	1.67	1.63	1.59	1.56	1.53
	Freon 12	5.18	5.03	4.89	4.77	4.65	4.53	4.42	4.32	4.22	4.13	4.05	3.95	3.88	3.81	3.74
CFM of refrigerant per minute per ton of refrigeration	Ammonia			20.4	14.9	11.1	8.40	6.45	5.00	3.96	3.13	2.52	2.04	1.695	1.38	1.14
	Propylene	27.1	20.2	15.7	12.0	9.18	7.30	5.85	4.74	3.84	3.11	2.53	2.12	1.80	1.51	1.28
	Propane	37.4	27.4	20.0	15.5	12.0	9.37	7.29	5.79	4.77	3.87	3.12	2.59	2.13	1.77	1.50
	Freon 12	59.9	43.2	31.7	23.7	18.0	13.9	10.8	8.52	6.79	5.47	4.44	3.63	3.00	2.50	2.09
Brake horsepower per ton of refrigeration	Ammonia			4.31	3.74	3.23	2.80	2.41	2.08	1.78	1.50	1.26	1.03	.835	.648	.483
	Propylene	5.00	4.47	3.96	3.51	3.10	2.69	2.35	2.06	1.74	1.46	1.20	1.00	.830	.647	.485
	Propane	4.98	4.39	3.87	3.43	3.03	2.67	2.32	2.03	1.75	1.49	1.24	1.01	.800	.622	.458
	Freon 12	5.70	4.98	4.33	3.79	3.31	2.86	2.47	2.14	1.83	1.55	1.30	1.06	.848	.668	.490
Condensed Liquid Temperature 125°F; Condenser Pressure in psia: Ammonia 303; Propylene 314; Propane 260; Freon 12 184																
Pounds of refrigerant per minute per ton of refrigeration	Ammonia			.492	.487	.483	.478	.474	.469	.466	.463	.460	.457	.454	.452	.450
	Propylene	2.67	2.58	2.50	2.42	2.35	2.28	2.22	2.16	2.11	2.06	2.01	1.97	1.93	1.89	1.86
	Propane	2.86	2.74	2.63	2.53	2.44	2.36	2.29	2.22	2.16	2.10	2.04	1.99	1.94	1.89	1.84
	Freon 12	6.42	6.19	5.98	5.80	5.61	5.45	5.28	5.14	5.00	4.87	4.75	4.64	4.53	4.43	4.33
CFM of refrigerant per ton of refrigeration	Ammonia			22.0	16.1	12.0	9.09	6.97	5.40	4.26	3.38	2.72	2.19	1.82	1.49	1.23
	Propylene	35.2	26.0	20.0	15.4	11.5	9.40	7.32	5.85	4.72	3.81	3.08	2.60	2.18	1.84	1.56
	Propane	50.0	35.6	25.8	19.7	15.4	11.8	9.16	7.20	5.94	4.79	3.79	3.14	2.63	2.19	1.80
	Freon 12	74.0	53.0	38.8	28.8	21.7	16.6	12.9	10.1	8.05	6.45	5.21	4.25	3.50	2.90	2.42
Brake horsepower per ton of refrigeration	Ammonia			5.68	4.96	4.38	3.81	3.33	2.92	2.54	2.19	1.90	1.63	1.38	1.16	.952
	Propylene	7.49	6.72	5.96	5.32	4.71	4.14	3.66	3.23	2.79	2.41	2.03	1.78	1.55	1.31	1.10
	Propane	7.47	6.60	5.85	5.18	4.60	4.06	3.59	3.18	2.81	2.43	2.07	1.77	1.50	1.25	1.03
	Freon 12	8.09	7.11	6.25	5.46	4.78	4.18	3.67	3.20	2.78	2.41	2.07	1.77	1.49	1.24	1.02

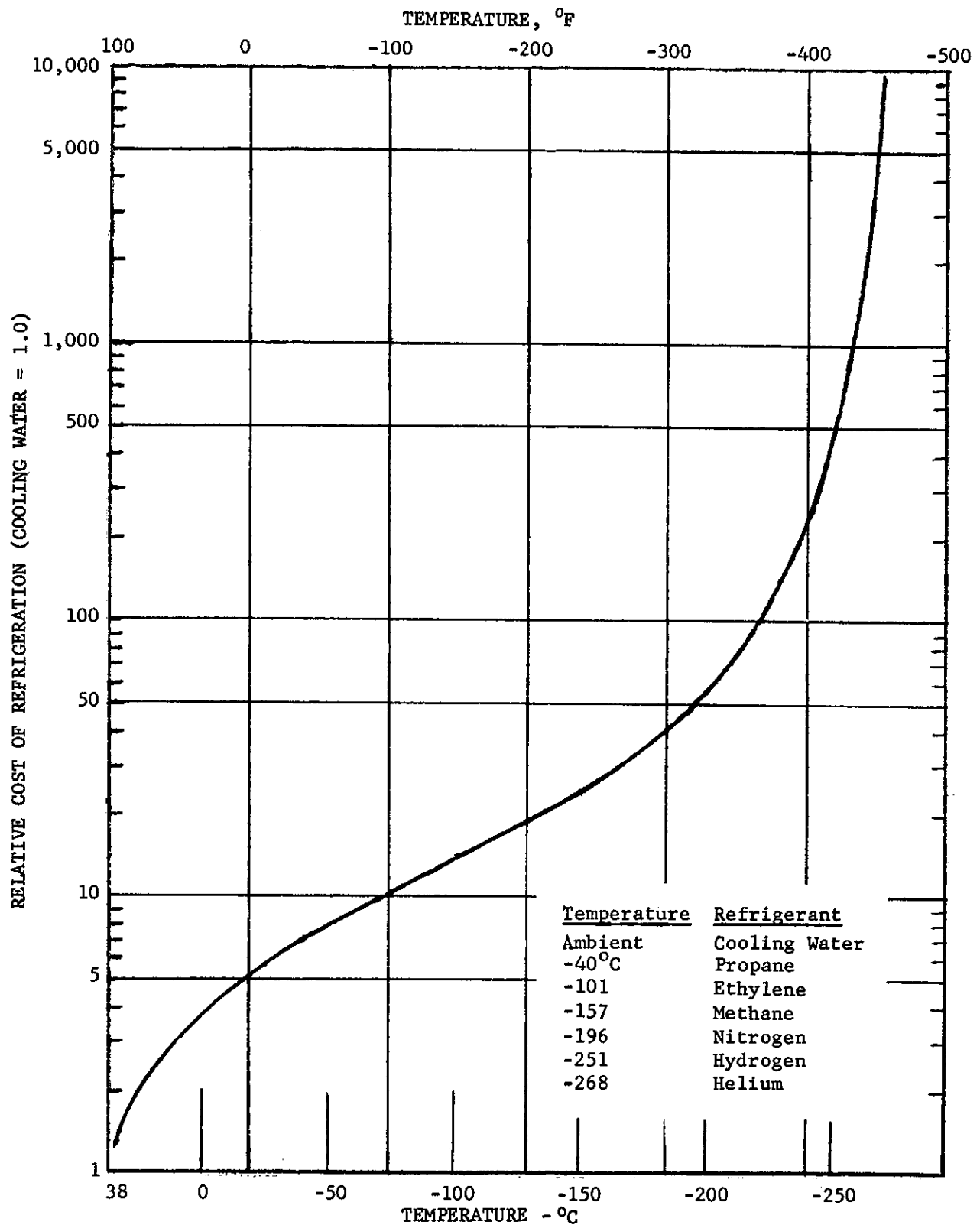
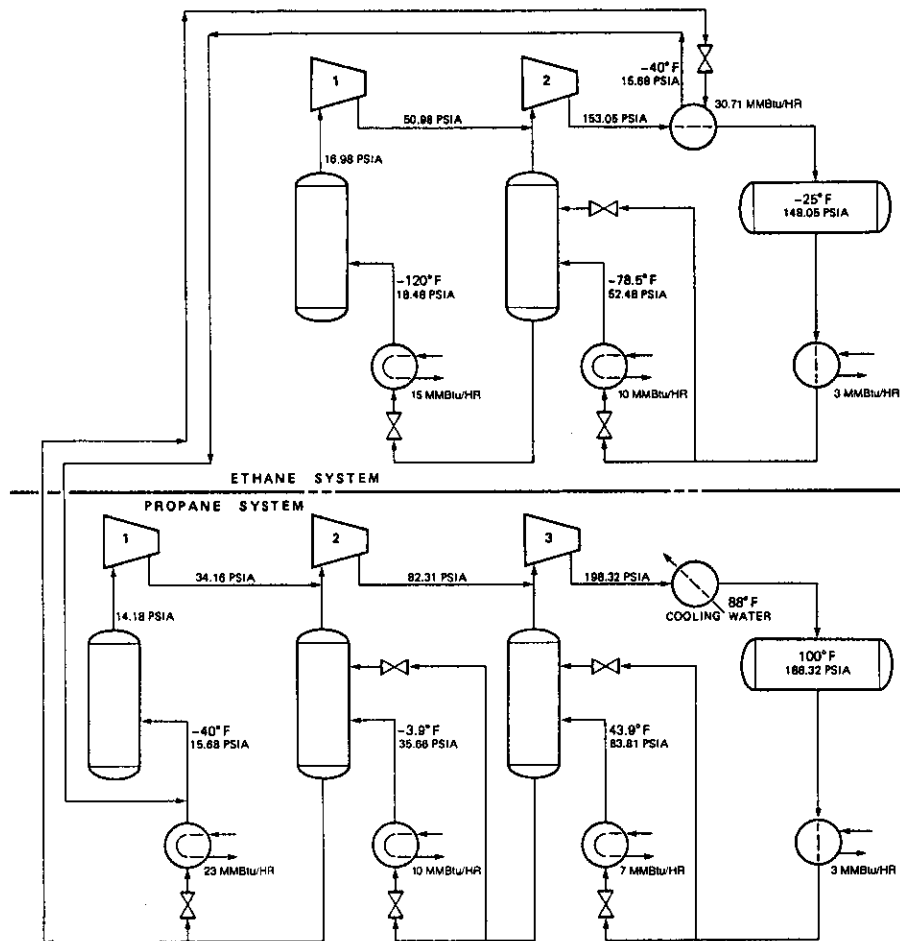


Figure 16.5 Relative Cost of Refrigeration Compared to the Cost of Cooling with Water at Ambient Conditions

Cascade Refrigeration

When refrigeration must be provided at very low temperatures $< -40^{\circ}\text{C}$ [-40°F], cascade refrigeration systems are sometimes used. Cascade systems employ more than one refrigerant and provide refrigeration at several levels. A propane cascade system is shown below.

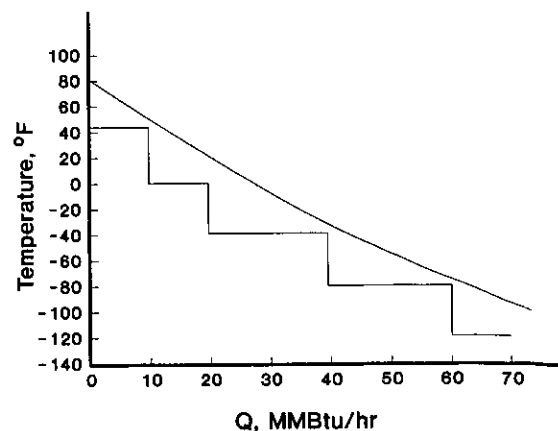


Courtesy
GPSA
Databook

In this system, refrigeration is provided at five levels

- 7°C [44°F]
- -20°C [-4°F]
- -40°C [-40°F]
- -61°C [-78°F]
- -84°C [-120°F]

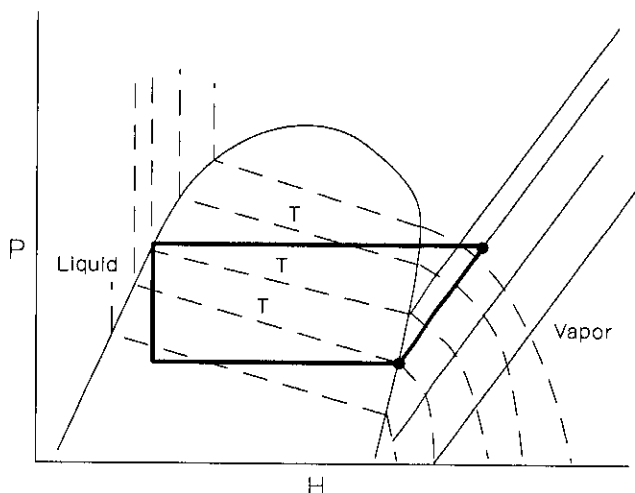
The propane at -40°C is used to condense the ethane refrigerant. All of the heat picked up in the process is ultimately rejected to the cooling water at the propane condenser. A hypothetical cooling curve for the process fluid has been developed to show the amounts and levels of refrigeration.



Cascade refrigeration systems are not common in gas processing. Low level refrigeration is typically provided using expansion or mixed refrigerants.

Mixed Refrigerants

An alternative to cascade refrigeration is to use a mixed refrigerant. Mixed refrigerants are a mixture of two or more components. The light components lower the evaporation temperature and the heavier components allow condensation at ambient temperatures. The evaporation process takes place over a temperature range rather than at a constant temperature as with pure component refrigerants. This is illustrated in the P-H diagram below for a mixed refrigerant.



The mixed refrigerant is blended so that its evaporation curve matches the cooling curve for the process fluid. Heat transfer occurs in a countercurrent exchanger, probably aluminum plate-fin, rather than a kettle-type chiller.

Mixed refrigerants have the advantage of better thermal efficiency, because refrigeration is always being provided at the warmest possible temperature.

The amount of equipment is also reduced compared to a cascade system. Disadvantages include a more complex design and the tendency for the heavier components to concentrate in the chiller unless the refrigerant is totally vaporized. The use of mixed refrigerants is very common in low temperature gas processing today and is standard practice in large LNG plants.

EXPANSION TURBINE REFRIGERATION

This has become a popular unit because of low relative cost and simplicity. The final temperature achieved at the outlet depends on the pressure ratio, the amount of liquid produced and the amount of actual work removed. In most cases to date, the maximum possible ΔT has been achieved, since cryogenic recovery was the goal – from ethane recovery through total liquefaction.

Actually, expanders offer applications at higher temperatures than the minimum achievable. If only propanes plus are to be recovered, there is no need to go to -70°C , even though one can achieve it. Excess methane and ethane are produced, which must be revaporized and maybe recompressed. It may be more economical to use the expander work output to produce electricity in some cases than for recompression. Expanders have become very popular for hydrocarbon dewpoint control where outlet temperatures are frequently greater than -30°C [-22°F].

One alternative is to send only part of the stream across the expander; the rest can be expanded across a valve. These parallel streams can then be commingled prior to separation. The proportion of gas through each parallel line can be controlled to produce the desired outlet temperature.

For an expander,

$$W_{\text{act}} = (E) (m) (h_2 - h_1)_{\text{theor}} \quad (16.4)$$

Where: E = efficiency (isentropic), expressed as a fraction
 m = mass flow rate
 h_2 = outlet enthalpy
 h_1 = inlet enthalpy

The first step is to find theoretical work by one of the methods of Chapter 15. The value of h_2 is found by trial-and-error by assuming a temperature so that $s_1 = s_2$. This involves assumption of the theoretical T_2 , running a flash and proceeding until the entropy check is obtained.

The true outlet temperature will be higher than the above theoretical temperature because actual work output is less than theoretical work output.

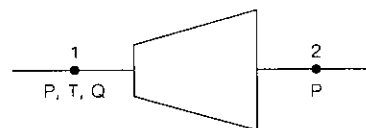
$$(h_2 - h_1)_{\text{act}} = (E) (h_2 - h_1)_{\text{theor}} \quad (16.5)$$

$$\text{So, } W_{\text{act}} = m (h_2 - h_1)_{\text{act}} \quad (16.6)$$

The efficiency may be estimated from Figure 16.8, but 80% is a good planning number.

The procedure for refrigeration is the same as the outlined above (and in Chapter 15). Once actual work has been found, one knows actual Δh . Since h_1 can be calculated from inlet composition, P_1 and T_1 , h_2 can be calculated. By trial-and-error one finds T_2 that corresponds to P_2 and the h_2 for the two-phase mixture leaving the expander.

As part of the process calculation, one will know (or specify) P_1 , P_2 , T_1 , the inlet volumetric flow rate (q_1), and composition. The general process is trial-and-error because enthalpy and entropy are explicit in T and P . The process is as follows:



1. From P_1 and T_1 , calculate h_1 and s_1 .
2. Assume a value of T_2 .
3. Run a flash calculation at the assumed T_2 and known P_2 to establish if liquid is formed.
4. Calculate h_2 and s_2 . (If the outlet is two-phase, these will be total stream values as shown in Chapter 8.)
5. If s_2 from (4) equals s_1 , you have assumed the right temperature. If not, repeat Steps 2-4 until $s_2 = s_1$.
6. Once $s_2 = s_1$, $h_2 - h_1 = \Delta h$ for the isentropic process.
7. Calculate Δh (actual): $\Delta h_{\text{act}} = (E)(\Delta h_{\text{theor}})$
8. Calculate actual power output using standard techniques discussed earlier for compressors.

This procedure is straight forward if one has a value for E , the isentropic efficiency. This will be discussed in a later section.

Since the above calculation process begins with the guess of a temperature, what represents a good first guess? Equation 16.7 may provide a good guessing value, even though it is based on ideal gases and assumes no liquid formation.

$$T_2 = T_1 + T_1 \left[\left(\frac{P_2}{P_1} \right)^m - 1 \right] (E) \quad (16.7)$$

Where: T = absolute temperature in consistent units
 P = absolute pressure in consistent units
 $m = (k - 1)/k$
 E = isentropic efficiency

If one has an enthalpy diagram of the fluid involved that includes P , T , and s , one can follow a constant entropy line to do the same thing.

Expander Performance

There are four basic types of expanders as shown in Figure 16.6. The impulse type is where all of the pressure drop occurs in the nozzles. In the reaction type, up to 50% of the drop occurs in the blades. The axial type is a series of blades while the radial type is more nearly a wheel.

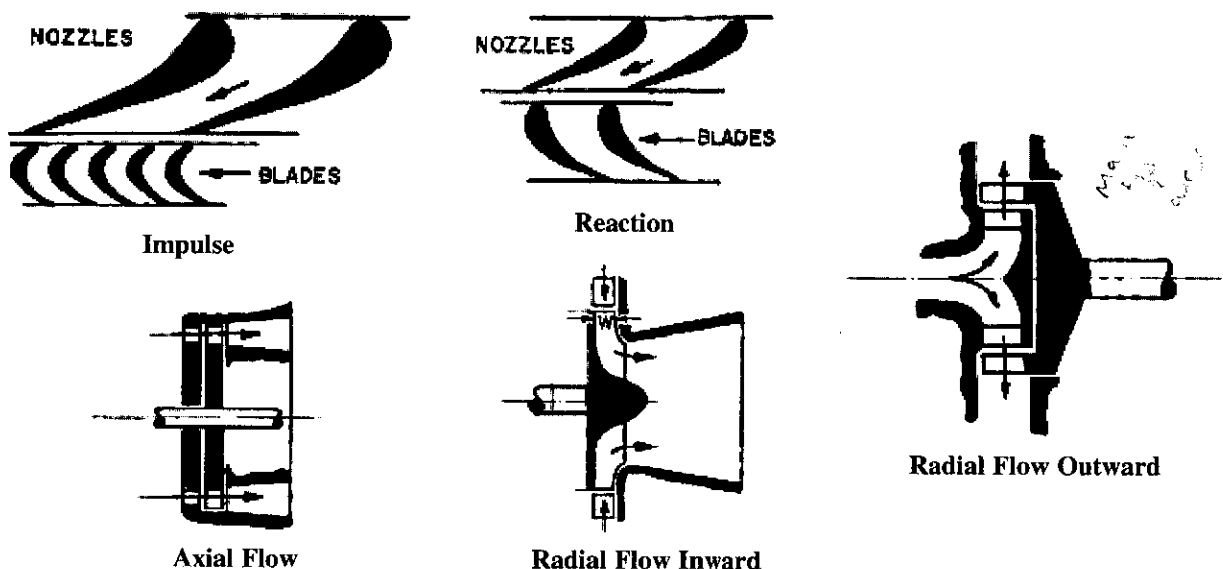


Figure 16.6 General Types of Expansion Turbines

There are many factors affecting choice. Is liquid condensed in the expander? Liquid possesses dynamic force that can damage the rotating element severely. More rugged construction is necessary with liquid present and it should be discharged without direct impingement on rotating elements. It is undesirable to have liquid or solids in the inlet stream.

The question of liquid formation is an important one. Some talk about a certain percentage of liquid "in the turbine." There is good reason to believe that a kind of nonequilibrium process occurs in the turbine proper at high flow rates. An equilibrium gas-liquid mixture only occurs in the outlet. So ... it is feasible to believe that most of the liquid forms just downstream of the wheel and does not interact with the wheel proper. Initially, we can expect any liquid formed to be of submicron size.

Although I cannot prove the above, there is much indirect evidence to support it. Many of our expanders in low temperature service are of the single-stage, radial inflow type. They are producing liquid in virtually all cases. If there were much liquid within the turbine wheel proper, the denser liquid would be

centrifuged outward, which would lead to lower efficiencies than we note because of flow distortions. Also, wheel erosion could be significant.

How much liquid can be handled? Some claim that the exit gas can contain up to 20% liquid. This can only be determined by experience. Certainly 10% is being accommodated and 20% is not unreasonable.

Figure 16.7 shows a cross-section of an expander directly driving a centrifugal compressor, one way to load it.

TURBOEXPANDER/COMPRESSOR
with totally enclosed seal system

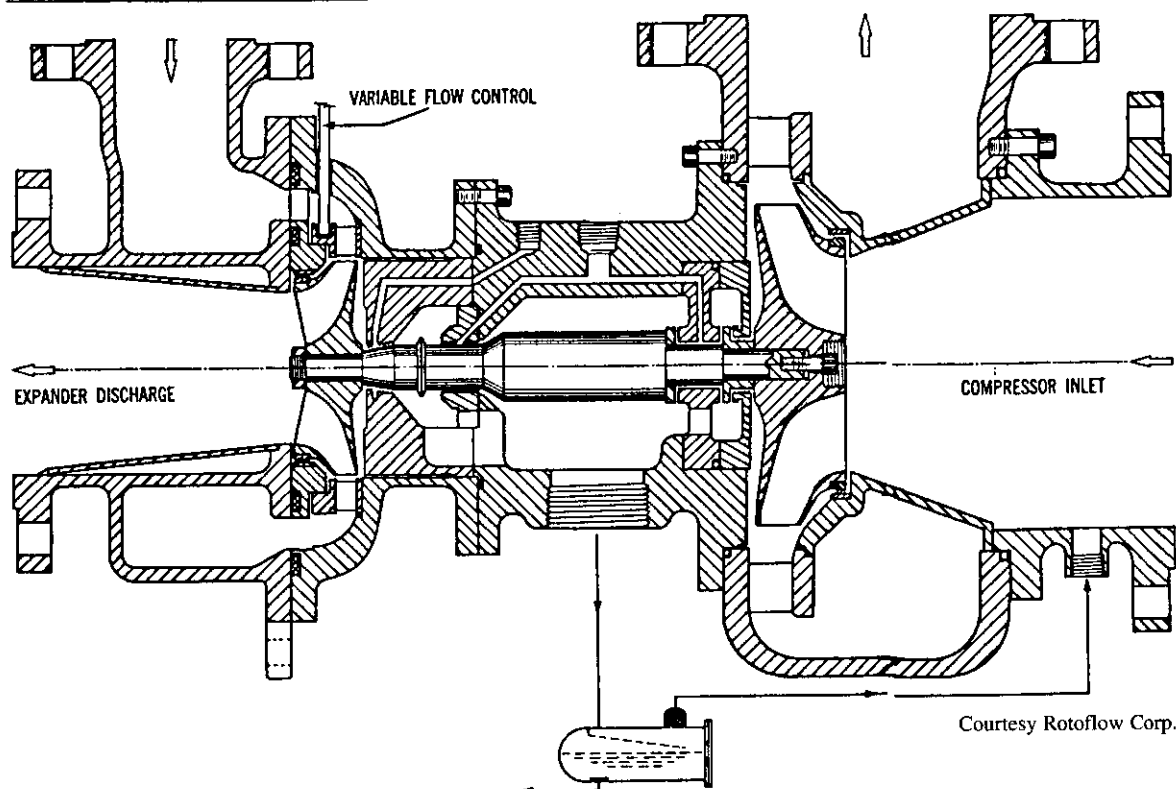


Figure 16.7 Efficiency of Expansion Turbines as a Function of Specific Speed

Estimation of Expander Efficiency

The best way to estimate efficiency is from actual performance data.

It is difficult to correlate efficiency data because many factors affect actual performance. As with rotating compressors, normal manufacturing tolerances can affect performance measurably. Erosion can alter the shape of a wheel and thus efficiency. The presence of liquids likewise may have a dramatic effect. For all of these reasons, there may be a significant error in the estimated efficiency.

In correlating efficiency data, one may use the basic similarity parameters governing all turbomachinery. It appears that correlation of E versus specific speed (N_s) is realistic. Figure 16.8 shows such a correlation based on data from several sources, including Reference 16.2.

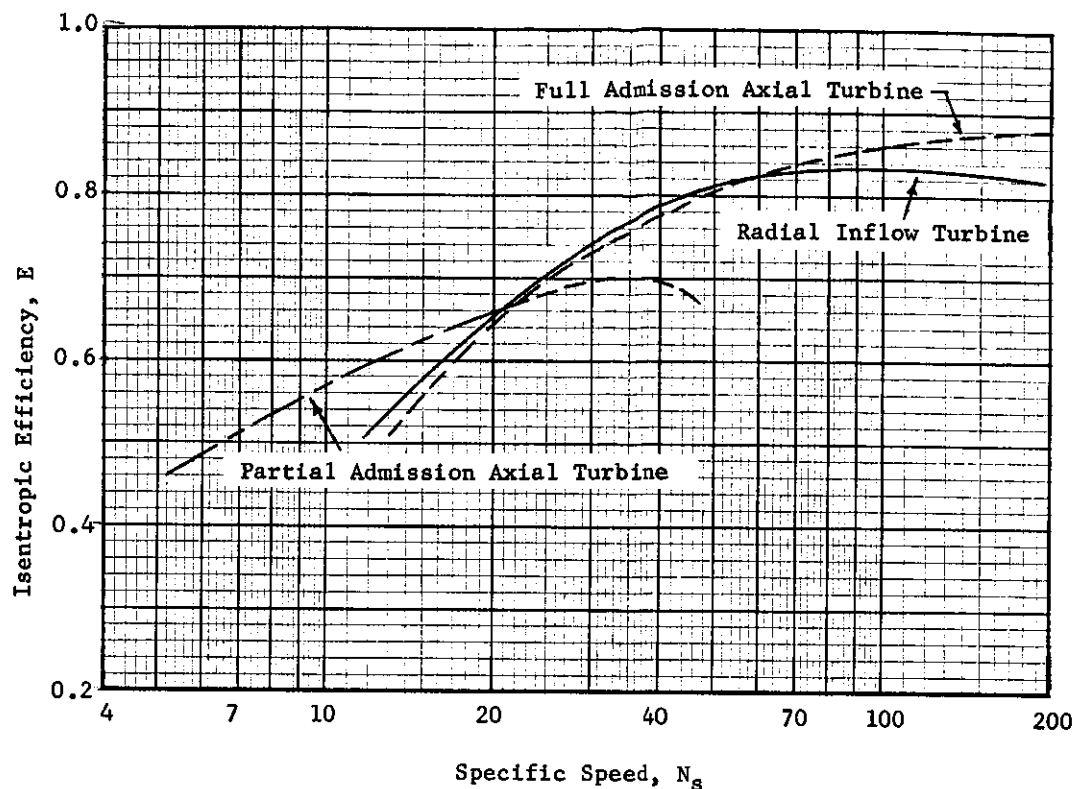


Figure 16.8 Efficiency of Expansion Turbines as a Function of Specific Speed

Specific speed is one of the criteria for determining performance. It was discussed previously in the centrifugal pump section. The basic equation for expanders is:

$$N_s = \frac{N (q_2)^{0.5}}{[(A)(\Delta h)]^{0.75}} \quad (16.8)$$

Where:

N	= shaft speed
Δh	= isentropic Δh
q_2	= turbine exhaust volume
A	= conversion factor

Metric	English
rpm	rpm
kJ/kg	Btu/lb
m ³ /s	ft ³ /sec
31	778

It is apparent from Figure 16.8 that specific speed should be above 70 to achieve maximum efficiency. For a radial inflow turbine, the optimum E is for a specific speed between 70 and 100. However, this is not always possible because of the limitation in sizes of expanders available and other parameters which affect performance.

Figure 16.8 is for expanders where the amount of liquid formed is minimal. How much does liquid affect efficiency? This depends on where, and how, it was formed. I have seen data on steam turbines where predicted efficiency was about double actual efficiency when too much liquid was formed in an axial turbine.

In the early planning stages, flow rates and gas compositions are rather inexact. Potential errors in both indirectly affect predicted efficiency. Some use a planning efficiency about 10% lower than that predicted by a correlation like Figure 16.7. I prefer a simulation calculation that recognizes the uncertainty and uses several values of E, to see what effect it has on those parts of the system depending on turbine performance. Obviously, E affects outlet temperature, gas-liquid ratio and the composition of both fluids which, in turn, affects all downstream equipment. I believe that picking one "magic" number is unrealistic.

Other Performance Criteria

In addition to specific speed there are other parameters affecting expander performance. Specific diameter is one of these, defined by the equation

$$d_s = \frac{(d) [(A)(\Delta h)]^{0.25}}{(q)^{0.5}} \quad (16.9)$$

Where: d = turbine diameter
 Δh = isentropic Δh
 q_2 = turbine outlet volume
 A = conversion factor

Metric	English
m	ft
kJ/kg	Btu/lb
m ³ /s	ft ³ /sec
31	778

The d_s shown above is a dimensionless number and would be the same when using both metric and English units. This was done to facilitate use of existing relationships between d_s and N_s . For natural gas processing the specific diameter should be approximately 1.2 to 1.4 to achieve maximum efficiency. This is a useful number for estimating impeller size and shaft speed.

Another parameter is the relative Mach number, which is simply the ratio of velocity at turbine discharge conditions to the speed of sound at the same conditions. Sometimes this ratio is replaced by the ratio (P_2/P_1).

The fourth parameter is a Reynolds number defined by the equation

$$Re = \frac{U_2 d}{\mu} \quad (16.10)$$

Where: U_2 = tip speed
 d = wheel diameter
 μ = kinematic viscosity

Any consistent set of units that makes Re dimensionless may be used.

The problem of the machinery specialist is to choose a machine that fits the system data supplied. Of these parameters, specific speed is the basic parameter of mutual concern between the various specialists involved.

VALVE EXPANSION REFRIGERATION

This is a relatively simple system used in production systems where a pressure drop is available and very low temperatures are not required. Figure 16.9 shows two versions of what are commonly called LTS or LTX systems.

PRESSURE DROP ACROSS VALVE

Pressure drop across a valve (choke) is an isenthalpic process, as noted previously. If no liquid forms, the following equation applies:

$$\mu = \left(\frac{\delta T}{\delta P} \right)_H = \frac{\left[T \left(\frac{\delta v}{\delta T} \right)_P - v \right]}{C_P} \quad (16.11)$$

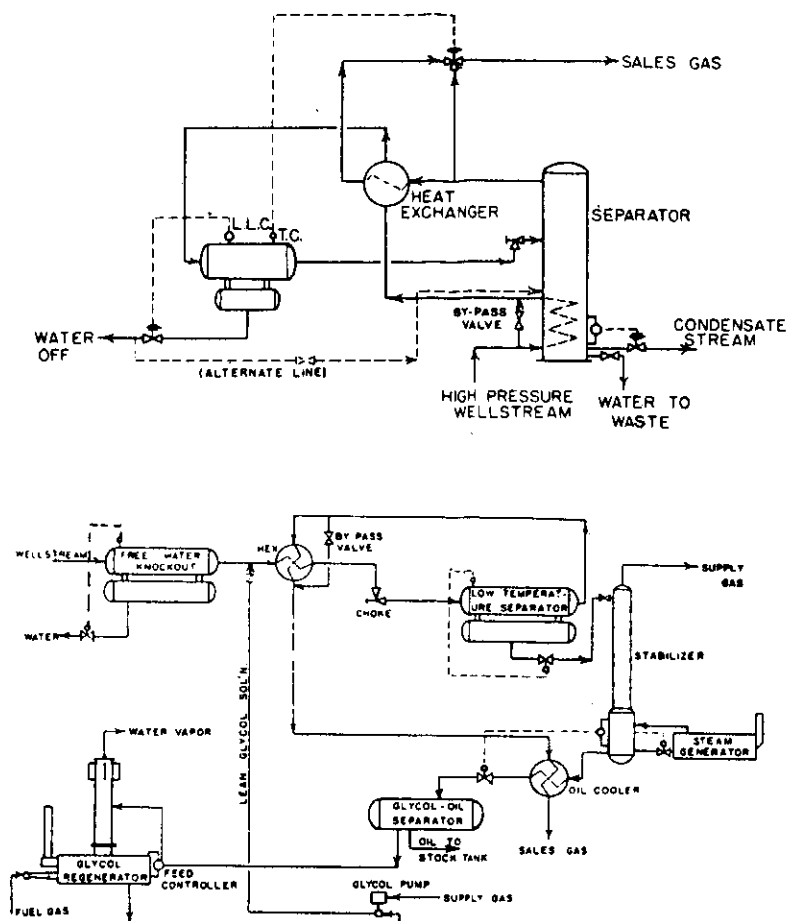
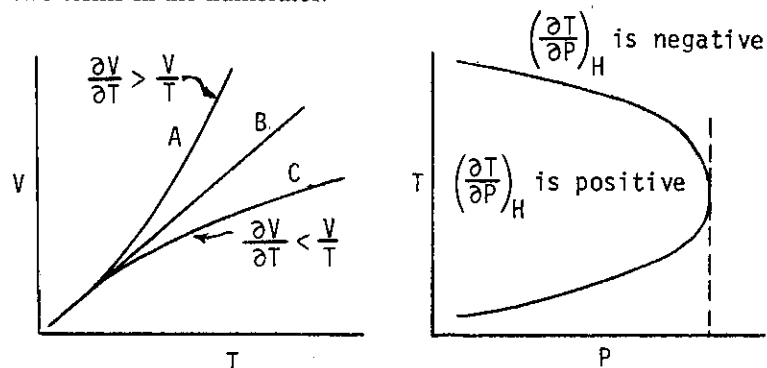


Figure 16.9 Two Types of LTS Systems Used in Production

The symbol " μ " is known as the Joule-Thomson coefficient. It is positive or negative, depending on the relative size of the two terms in the numerator.



Curve A below shows a case where the instantaneous slope is greater than the average slope. Therefore, the gas will cool on expansion. The curve C gas is just the opposite and will heat on expansion. Curve B is for an ideal gas, which will not change temperature on expansion.

Many gases exhibit a characteristic wherein the slope of the V-T curve changes sign. The temperature at which the slope changes sign ($\mu = 0$) is known as the *inversion temperature*. The right-hand plot above shows inversion temperature versus pressure. The shape shown is general for all actual gases. Outside the curve, the gas represented would heat upon expansion. Inside, it cools on expansion.

Because of the location of the curve, hydrogen heats on expansion at normal pressures, whereas most light hydrocarbons cool. At very high pressures, of the order of 60 MPa [8700 psia], many naturally occurring hydrocarbon gases heat on expansion.

Curves which show the temperature drop expected for a given pressure drop across a choke are only applicable if no liquid forms on such expansion.

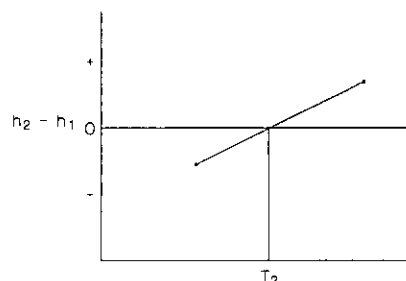
Liquid Condensation Across a Valve

Across a valve the First Law of Thermodynamics reduces to $h_1 = h_2$. No work is possible and the process is almost adiabatic. The amount of heat transfer across a valve body is poor, and the gas is in it for only a short time. The calculation is inherently trial-and-error.



1. Calculate the total enthalpy of the feed stream at P_1 and T_1 . If it is a two-phase stream, the total enthalpy is found by adding that of the liquid and vapor phases.
2. Assume the unknown temperature T_2 .
3. Make a flash calculation at P_2 and T_2 to find relative amount and analysis of each phase.
4. Find the total enthalpy at Point Two from the above flash and the assumed T_2 .
5. If $h_2 = h_1$, you assumed the right temperature. If not, repeats Steps 2-5 until the h 's are equal, within the desired limits of accuracy.

Since the above is very tedious, a reasonably good answer usually can be obtained by assuming two different temperatures and plotting them on the following type of figure.



A straight line between the $h_2 - h_1$, found for two assumed temperatures, are connected by a straight line. The intersection at $h_2 - h_1 = 0$ gives approximate true T_2 .

For the systems shown in Figure 16.9, the inlet gas stream is cooled by the exit separated gas before going to the choke. All one knows are the inlet conditions and composition and the sales gas limitations. These fix the pressure drop across the heat exchanger and choke in series. The drop across the former should not exceed 100 kPa. The following type of procedure is needed.

1. Assume temperature of gas downstream from HEX.
2. Run flash calculation at this temperature and inlet pressure, minus 70 kPa [10 psia].
3. Determine enthalpy of total stream at this point from composition in Step 2.
4. Use the previous procedure for a choke to find the temperature in the low temperature separator.
5. Run flash at separator conditions.
6. Find the enthalpy of the vapor leaving the separator.
7. The ΔH of the sales gas must equal the ΔH of the inlet gas across the gas-gas HEX. If this is not found to be true, Steps 1-6 must be repeated.
8. Once Step 7 is satisfied, the heat exchanger may be found by conventional heat transfer principles.

Note: A flash calculation is needed on the inlet gas to the HEX if it is two-phase and composition and relative quantity of each phase is not known.

The above procedure illustrates the general conditions that must be satisfied for all systems where expansion across a valve is involved.

Expansion across a valve may be the proper choice over an expander but the temperature drop is less and no useful work is produced.

APPLICATIONS OF REFRIGERATION

There are many ways in which refrigeration is applied. Figure 16.10 shows an application for refrigerated propane storage.

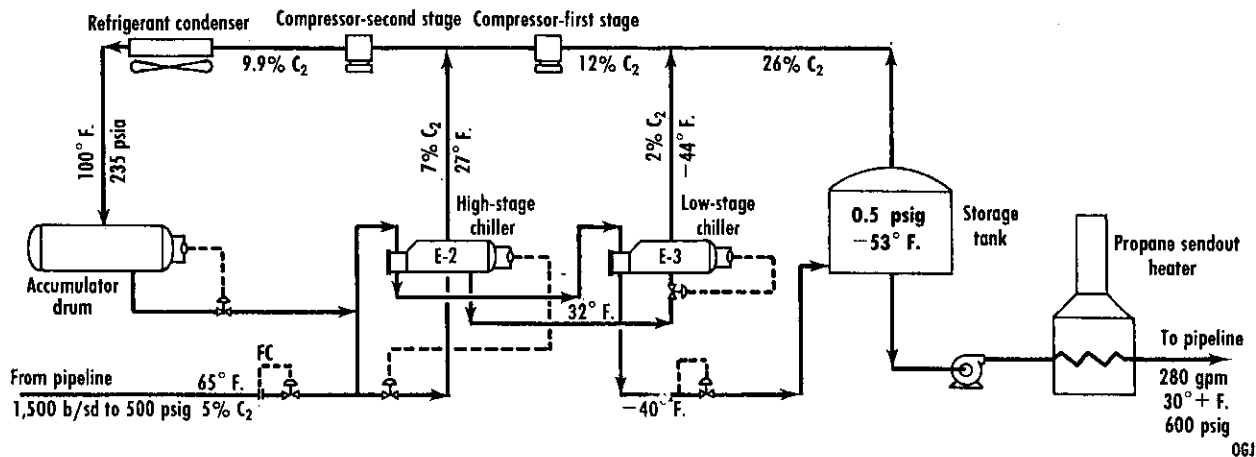


Figure 16.10 Application of Compression Refrigeration to Propane Storage

Figure 16.11 shows a flowsheet for a very simple refrigeration plant with a stabilizer used for a field installation. Note that a salt bath heater is used and to simplify the system there is no reflux on the stabilizer.

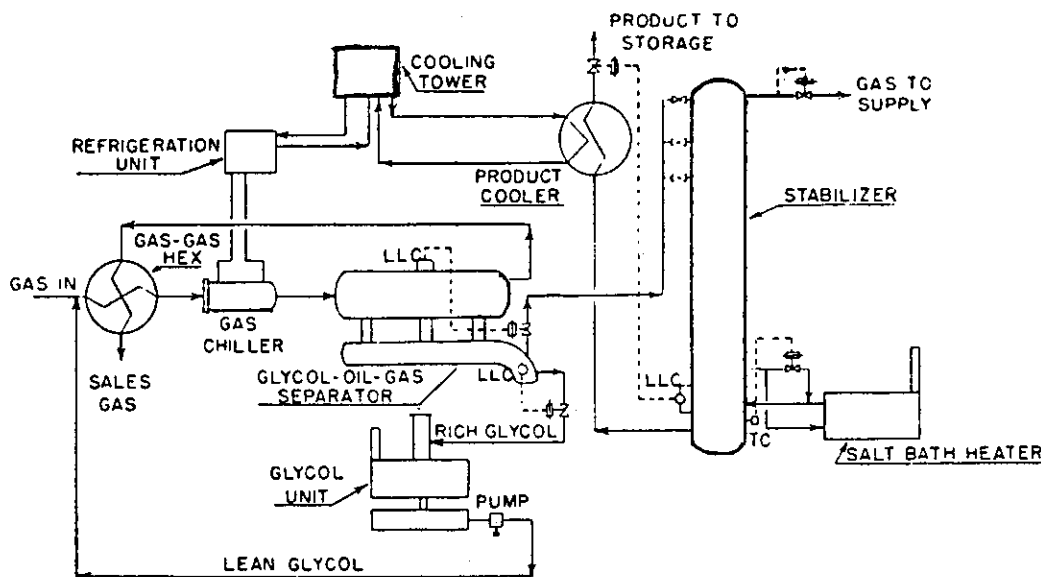
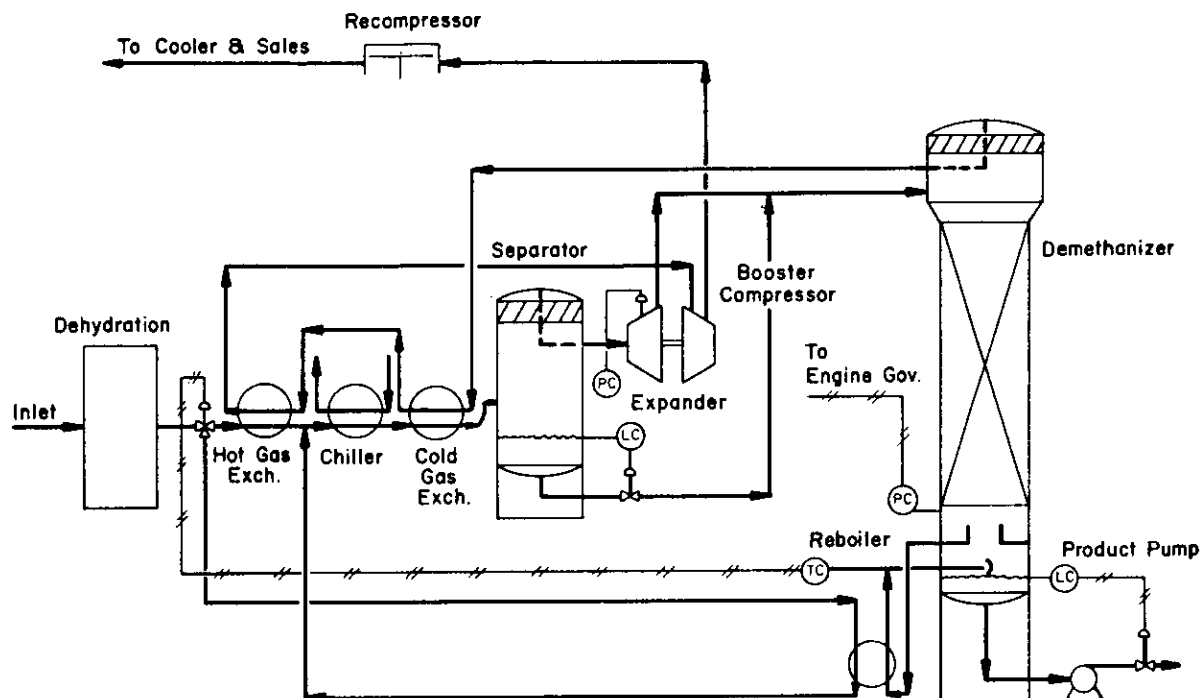


Figure 16.11 Flow sheet for Simple Refrigeration Plant with Stabilizer

Figure 16.12 shows a typical turboexpander process used to process gas with a C₂+ content of 200-700 L/106std m³ [1.5 to 5 gpm]. For leaner gas the process is essentially identical except the chiller is not required. Typical inlet pressures are 4000-8000 kPa, with expansion ratios of 3 to 4:1 across the turboexpander. Expander outlet temperatures vary with recovery objectives and inlet CO₂ content but frequently range from -80 to -120°C [-112 to -184°F].



Typical for 1.5 to 5 gpm of C₃

Figure 16.12 Flow Sheet for Simple Expander Plant

For deep ethane recovery (70%+) the inlet gas usually provides most, if not all, of the reboiler heat. Although not indicated in Figure 16.12, side reboilers are a standard design. These are used to control vapor loading in the demethanizer, prevent CO₂ freezing and to optimize thermal efficiency of the heat exchanger train. The demethanizer column is frequently non-refluxed with a top tray feed. The top of the demethanizer acts as a low pressure separator to avoid cryogenic pumping of condensed hydrocarbons to the top of the tower.

In older plants the heat exchangers are shell-and-tube design; however, in newer facilities plate-fin exchangers are very popular. The dehydration process is almost always molecular sieve, although some designs are based on glycol absorption plus methanol injection. The primary energy input to the facility is at the recompressor. Compression ratios at the booster compressor generally vary from 1.3 to 1.5.

Flow through the expander is controlled by a set of adjustable inlet guide vanes pictured below. These guide vanes are usually actuated by a pressure controller controlling either upstream or downstream pressure.



Courtesy Rotoflow Corp.

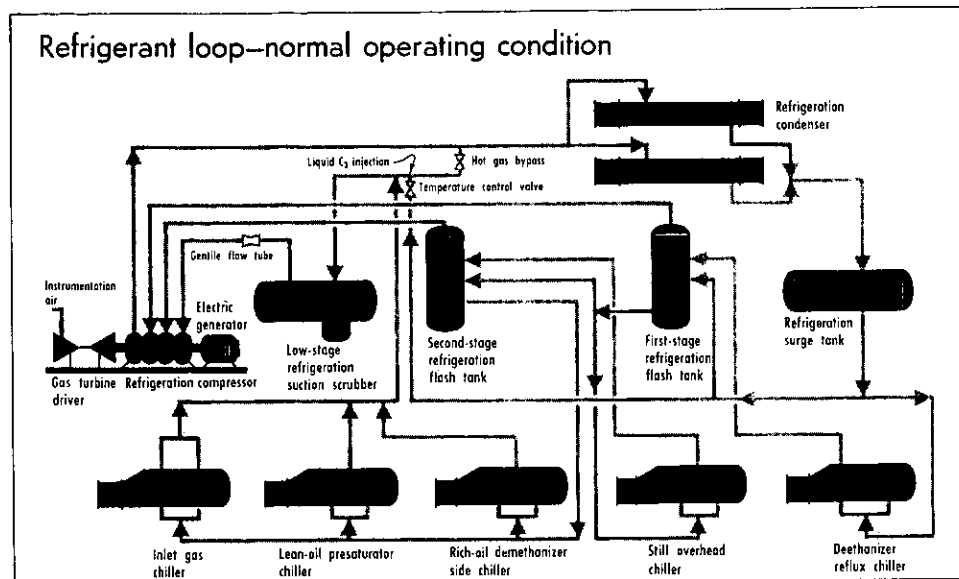
Expander plants are probably the most common type of gas processing facility used today. They combine high recovery levels with low capital cost, simplicity and ease of operation. They are less attractive on very rich gas streams (e.g., low pressure solution gas) or where the light NGL produces, C_2 and C_3 , are not marketable.

Figure 16.13 is for a plant using a gas turbine to drive a centrifugal propane compressor and an a.c. generator in tandem. This unit provides all refrigeration, electricity, and instrument air needed. Waste heat from the turbine supplies most of the process energy. The plant feedstock is from off-shore, transported through a two-phase pipeline. The design inlet rate is $5.2 (E+06)$ std m^3/d with recovery of about $1190 m^3/d$ of crude/condensate and $875 m^3/d$ of light hydrocarbons. Refrigerated absorption oil is used to recover about 55% of the ethane and almost all of the propane and heavier. Refrigeration is to minus $40^\circ C$. Shown also in Figure 16.13 is a comparison of the scheme utilized with use of steam throughout for both power and process.

The following equipment has been used:

- 1 – 6900 kW site rated, single shaft, simple-cycle gas turbine (started by expanding gas through a 900 kW turbine)
- 1 – 71 500 000 kJ/h waste heat unit, heating 240 000 kg oil per hour from $182-288^\circ C$. (Waste heat is supplemented by burners to heat the turbine exhaust from $427-540^\circ C$.) The overall thermal efficiency with supplemental firing is 63%.
- 1 – Seven impeller, horizontal split, cast steel case, centrifugal compressor rated at 4580 kW at 600 rpm. Suction and interstage pressure are 82, 210 and 520 kPa. Discharge is at 1610 kPa and $80^\circ C$. The economizer connections are between the third and fifth stages. Dry desiccant drying is used.

Figure 16.14 shows one arrangement for recovering liquid from oil treater and stock tank vapors.



Original estimate cost summary

Variable main equipment	A One 8,700-hp (NEMA) gas turbine	B Two 4,250-hp (NEMA) gas turbines	C One 8,000-hp steam turbine	D Two 4,000-hp steam turbines
Driver(s) and compressor	794,000	1,116,000	322,000	430,000
Waste-heat oil heater	162,000	180,000		
600-lb steam boilers (three 90,000-lb/hr)			365,000	365,000
Cooling tower	103,000	169,000	169,000	169,000
Balance main equipment	185,000	185,000	308,000	308,000
Total installed	1,244,000	1,584,000	1,164,000	1,272,000
Additional investment Over one gas turbine	2,631,000	3,331,000	2,827,000	3,041,000
		700,000	156,000	410,000

This reveals the economics of size and that the gas turbine scheme for both power and process heat is attractive only when using only one gas turbine unit, instead of splitting the service between two units.

Downtime comparison—gas turbine vs. two steam turbines (assuming no loss with two steam turbines)

1. Incremental investment—\$410,000				
2. Value of production at 1/4 = \$8,000/day				
Incremental expenses—two steam turbines over one gas turbine				
Assumed gas-turbine shutdown days	None	5 days/year	10 days/year	20 days/year
Gas turbine	\$390,000	\$425,500	\$461,000	\$531,000
Steam turbine	476,000	476,000	476,000	476,000
Incremental expense two steam turbines over one gas turbine	86,000	50,500	15,000	(55,000)
Steam turbine incremental investment = \$410,000				
Gas turbine incremental expenses (20 day/year downtime) = 55,000				
= 7.45 years				

Conclusion: Shutdown time for gas turbine would have to exceed 20 days/year before two steam turbines would become more attractive. Reliability of gas turbine was considered between 97 to 99%; therefore, 4 to 10 days/year of gas turbine downtime would be expected.

Generation vs. purchased power (original estimate)

Gas turbine for refrigerant load and generator for total electrical load (10,000 hp) vs. gas turbine (7,000 hp) for refrigerant load and all electrical load purchased.

Purchased electrical power: 2,272 kw

Costs: at 0.7 cents/kwh = $2272 \times 24 \times 365 \times 0.007 = \$139,000/\text{year}$

Costs: at 1.0 cents/kwh = $2272 \times 24 \times 365 \times 0.01 = \$199,000/\text{year}$

Operating and maintenance: \$5.50/hp-year for gas turbine

\$2.00/hp-year for generator

3,000 hp = \$23,000/year incremental for generator

Incremental profitability: Fuel current basis

Power cost	Purchased electrical	Generator O & M	Fuel	Net cost difference	Taxes, depreciation, etc.	Net income
At 0.7 cents/kwh	\$139,000	(\$23,000)	(\$26,000)	\$90,000	\$34,000	\$56,000
At 1.0 cents/kwh	\$199,000	(\$23,000)	(\$26,000)	\$150,000	\$63,000	\$87,000

Assumed purchased power lost 0.7 cents/kwh

Net income over life (15 years) after taxes and fuel = \$840,000

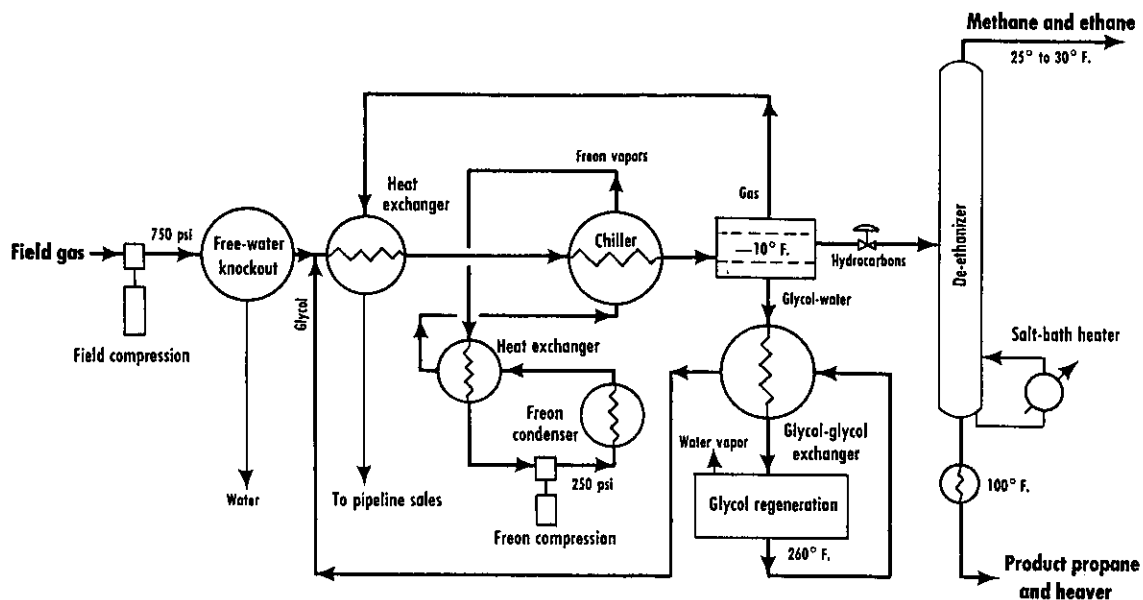
Investment for generator (incremental turbine

hp, plus generator, gear, and exciter) = \$390,000

Incremental profit after taxes and fuel = \$450,000

The capital investment required for the proposed electric-power generating facilities were economically justified at a price for purchased power as low as 0.7 cents/kwh-hr.

Figure 16.13 Flow Sheet and Data for a Compression Refrigeration Plant^(16.5)



Inlet gas flow rate: about 700 Mscf/d
 Product: about 3000 U.S. gallons per day
 Glycol injection is used.
 Freon 502 is used as refrigerant.

Figure 16.14 Refrigeration Plant for Recovering Liquid from Oil Treater and Tank Vapors

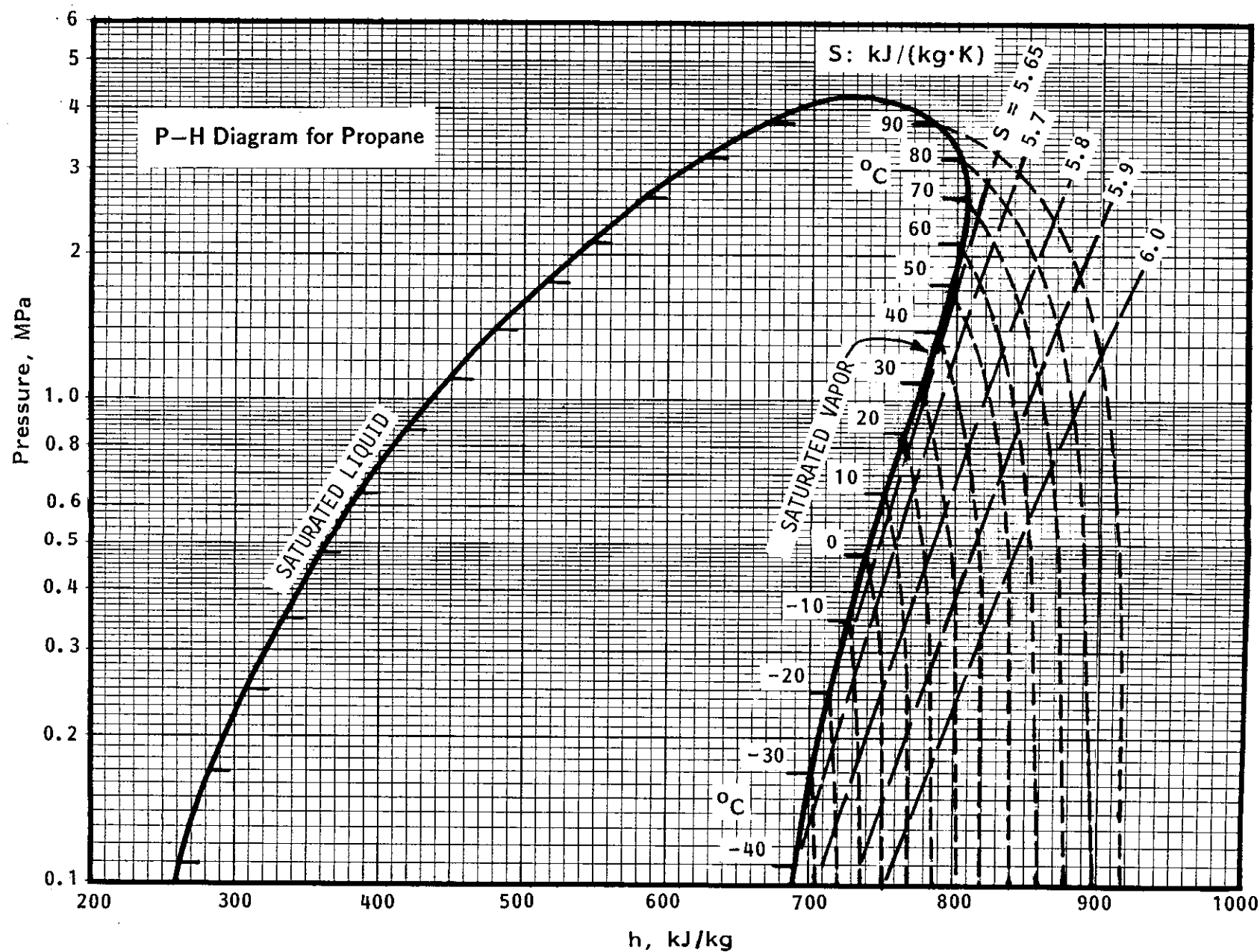
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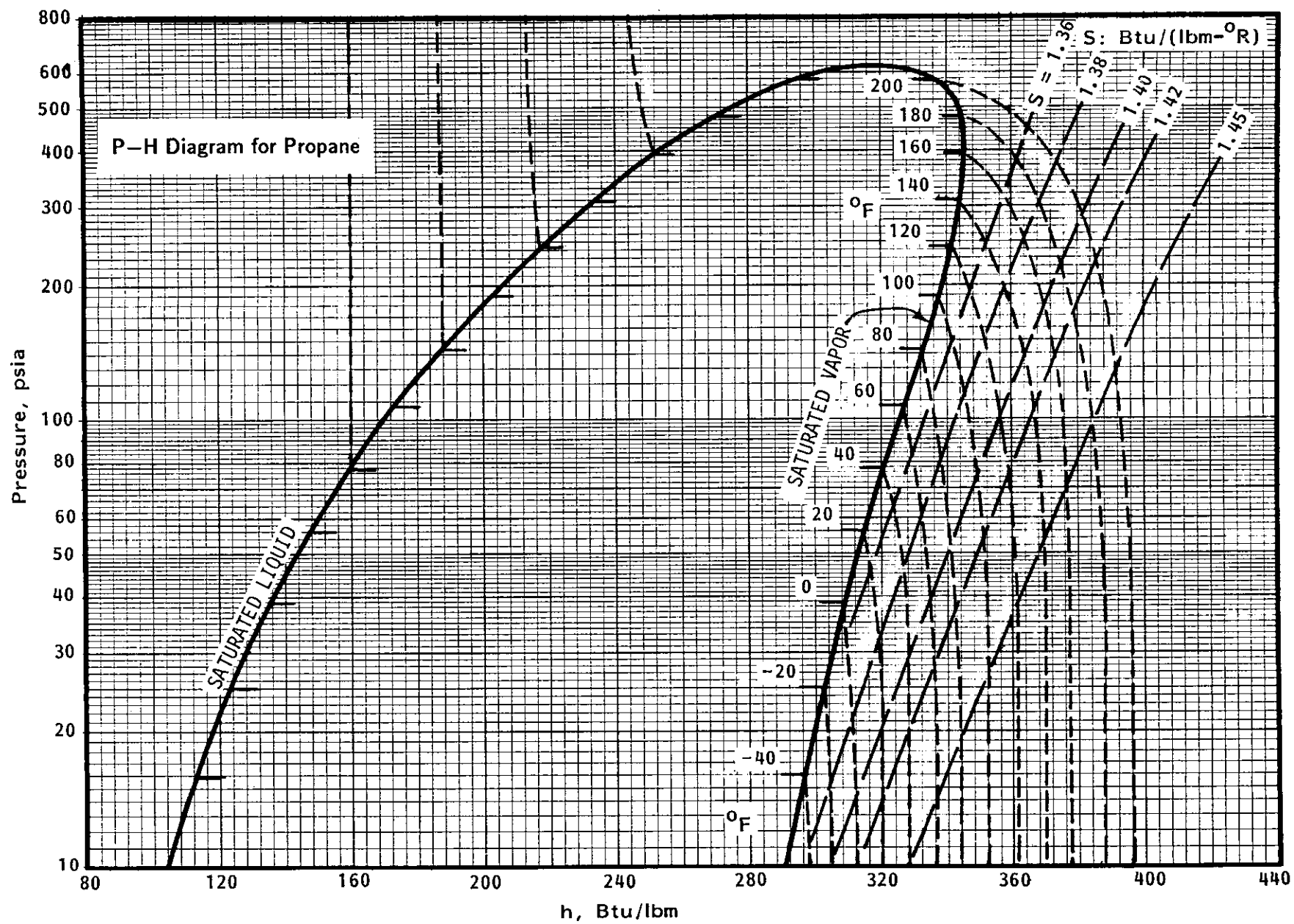
- 16.1 Briley, G. C., *Hydr. Proc.* (May 1976), p. 173.
- 16.2 Linhardt, H. D., *LNG/Cryogenics* (Feb. 1973), p. 7.
- 16.3 Bleakley, W. B., *Oil Gas J.* (April 3, 1967), p. 236.
- 16.4 Herrin, J. P., *Ibid.* (June 20, 1966), p. 181.
- 16.5 Davis, C. D., *Ibid.* (Mar. 25, 1968), p. 132.

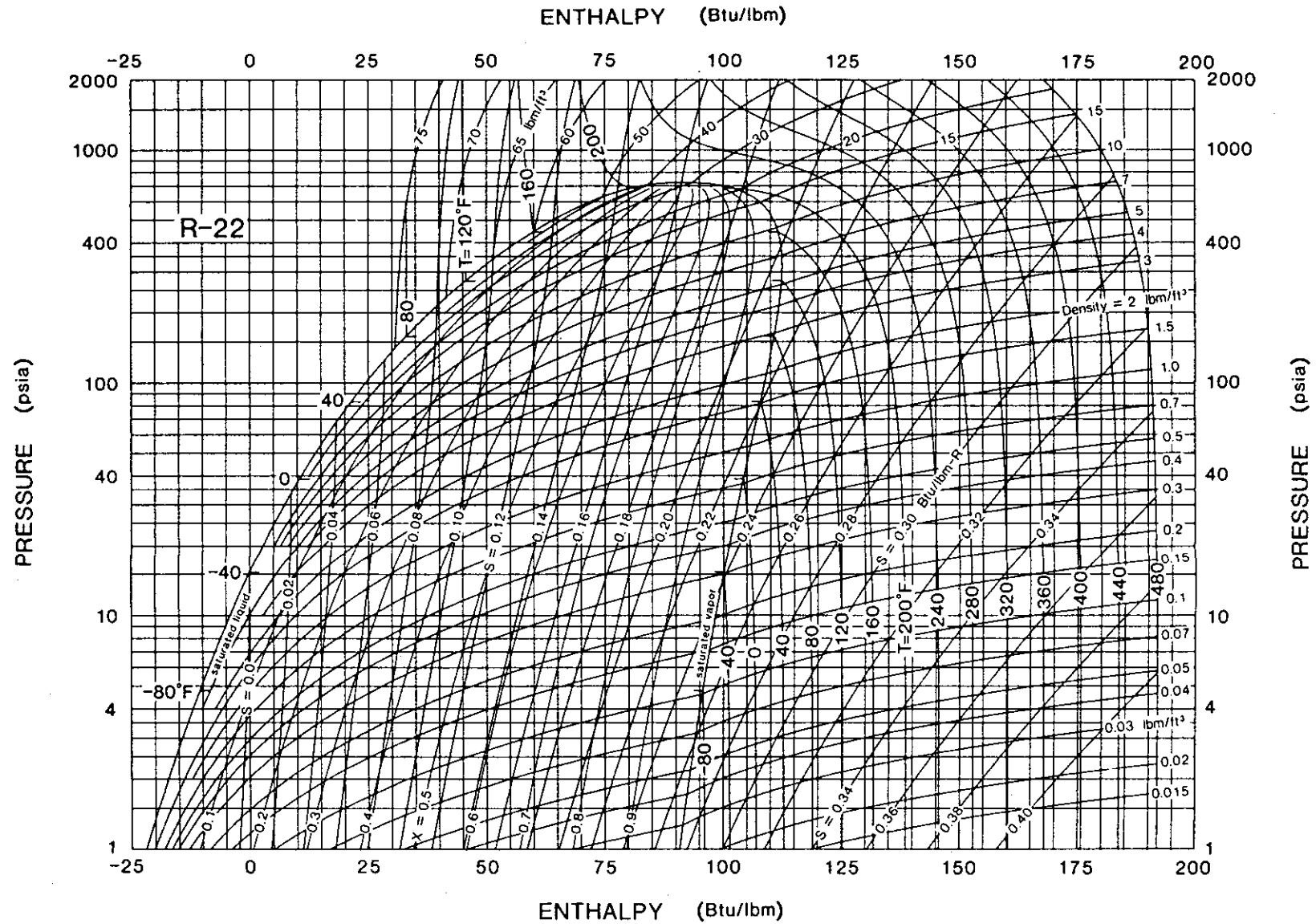
APPENDIX 16A

PROPERTIES OF COMMON REFRIGERANTS

(See also Appendix B at the back of this book.)





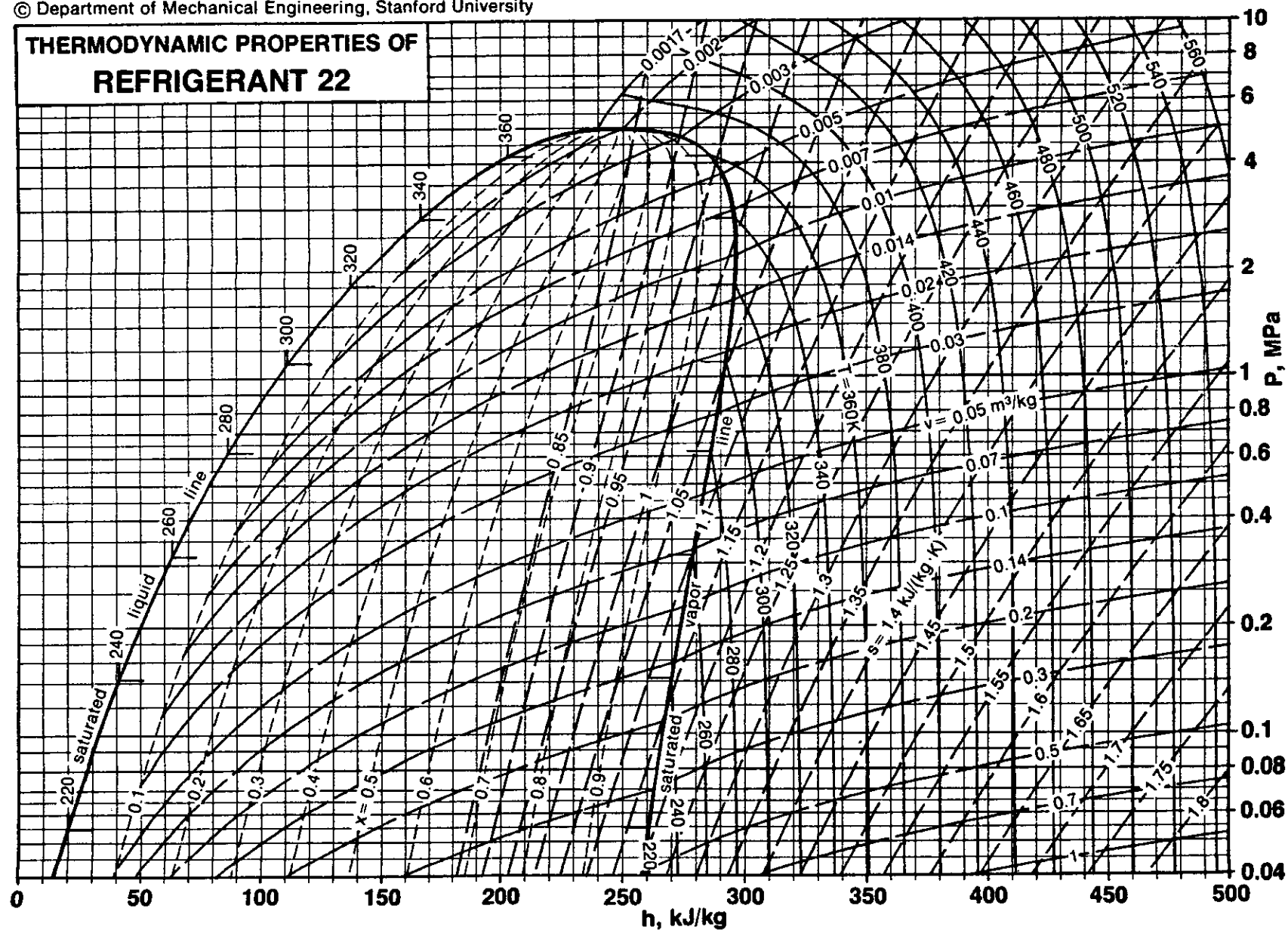


Pressure-Enthalpy Diagram for Refrigerant 22

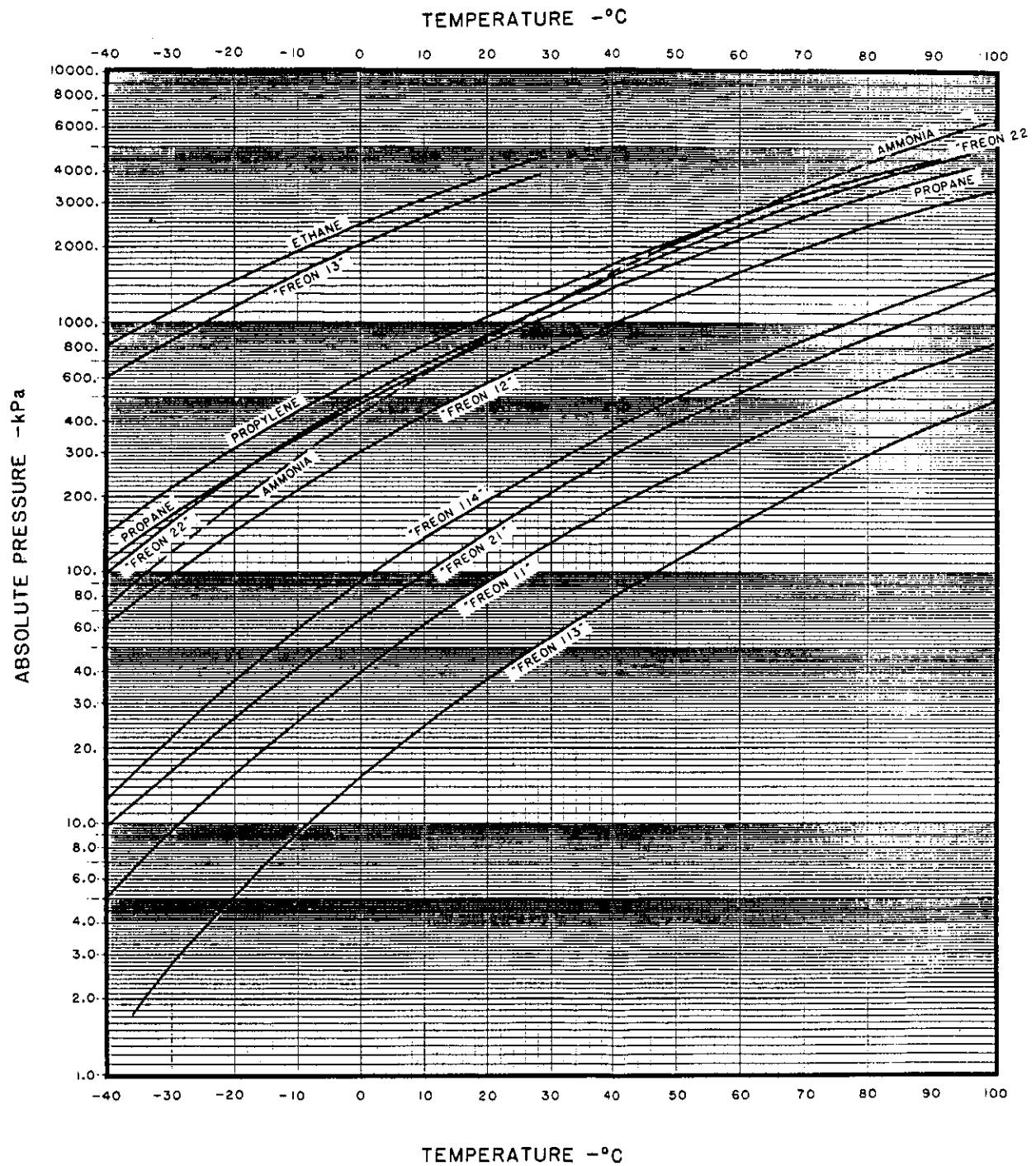
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THERMODYNAMIC PROPERTIES OF REFRIGERANT 22



PRESSURE - TEMPERATURE RELATIONSHIPS REFRIGERANTS



Comparative Data of Refrigerants

	Refrigerant Number (ARI Designation)					
	F-11	F-12	F-22	F-113	F-114	F-500
Chemical Formula	CCL_3F	CCl_2F_2	CHClF_2	$\text{CCl}_2\text{F}-\text{CClF}_2$	$\text{C}_2\text{Cl}_2\text{F}_4$	73.8% CCl_2F_2 26.2% CH_3CHF_2
Molecular Wt	137.38	120.93	85.48	187.39	170.93	99.29
Boiling pt. @ 101.3 kPa	23.7	-29.8	40.8	47.6	3.6	-33.3
Critical temperature, °C	197.8	112	96	214.1	145.7	122.8
Critical Pressure, MPa	4.38	4.11	4.94	3.41	3.27	4.35
Specific heats kJ/kg·°C of liquid @ 30°C	0.920	0.983	1.402	0.912	0.996	1.255
C_p of vapor @ 16°C, 101.3 kPa		0.611	0.623		0.653	0.715
C_v of vapor @ 16°C, 101.3 kPa		0.544	0.531		0.607	0.632
Ratio $C_p/C_v = K$ (30°C, 101.3 kPa)	1.11	1.14	1.18	1.12	1.09	1.13
Ratio of Specific heats liquid, 46°C vapor C_p , 40°C sat. press.	2.04	1.55	2.14	1.47	1.59	1.77
Odor	Ethereal, odorless when mixed with air	Same	Same	Same	Same	Same
Viscosity Centipoise Saturated liquid @ 35°C	0.3893	0.2463	0.2253	0.5845	0.3420	0.2150
@ 41°C	0.3723	0.2395	0.2207	0.5472	0.3272	0.2100
Vapor @ 101.3 kPa and -1°C	0.0101	0.0118	0.0120	0.0097	0.0108	
and 4°C	0.0103	0.0119	0.0122	0.0098	0.0109	
and 10°C	0.0105	0.0121	0.0124	0.0100	0.0111	
Thermal Conductivity ⁽¹⁾ Saturated liquid @ 35°C	0.371	0.300	0.357	0.319	0.271	
@ 41°C	0.362	0.292	0.345	0.312	0.262	
Vapor @ 101.3 kPa and -1°C	0.028	0.029	0.037	0.023	0.035	
and 4°C	0.029	0.031	0.038	0.024	0.036	
and 10°C	0.029	0.032	0.039	0.025	0.037	

⁽¹⁾ k in kJ·m/h·m²·°C

PROPERTIES OF COMMON REFRIGERANTS

Comparative Data of Refrigerants

	Refrigerant Number (ARI Designation)					
	F-11	F-12	F-22	F-113	F-114	F-500
Chemical Name	Trichloromono-fluoromethane	Dichlorodi-fluoromethane	Monochlorodi-fluoromethane	Trichlorotri-fluoroethane	Dichlorotetra-fluoroethane	Azeotrope of Dichlorodi-fluoromethane and Difluoroethane
Chemical Formula	CCl_3F	CCl_2F_2	CHClF_2	$\text{CCl}_2\text{F}-\text{CClF}_2$	$\text{C}_2\text{Cl}_2\text{F}_4$	73.8% CCl_2F_2 26.2% CH_3CHF_2
Molecular wt	137.38	120.93	86.48	187.39	170.93	99.29
Gas Constant, R (ft-lb/lb-R)	11.25	12.78	17.87	8.25	9.04	15.57
Boiling point at 1 atm (F)	74.7	-21.62	-41.4	117.6	38.4	-28.0
Freezing point at 1 atm (F)	-168	-252	-256	-31	-137	-254
Critical temperature (F)	388.0	233.6	204.8	417.4	294.3	221.1
Critical pressure (psia)	635.0	597.0	716.0	495.0	474.0	631.0
Specific heat of liquid	86°F	0.220	0.235	0.335	0.218	0.238
Specific heat of vapor, C_p 60°F at 1 atm	*	0.146	0.149	*	0.156	0.171
Specific heat of vapor, C_v 60°F at 1 atm	*	0.130	0.127	*	0.145	0.151
Ratio $C_p/C_v = K$ (86°F at 1 atm)	1.11	1.14	1.18	1.12	1.09	1.13
Ratio of Specific Heats Liquid, 105°F Vapor, C_p , 40°F sat. press.	2.04	1.55	2.14	1.47	1.59	1.77
Liquid heat (ft), 1 psi at 105°F	1.61	1.84	2.04	1.51	1.65	2.10
Saturation pressure (psia) @ -50°F	0.52	7.12	11.74	*	1.35	*
@ 0 °F	2.55	23.85	38.79	0.84	5.96	27.96
@ 40°F	7.03	51.67	83.72	2.66	15.22	60.94
@ 105°F	25.7	141.25	227.65	11.58	50.29	167.85
Net refrigerating effect (Btu/lb) 40-105°F (no subcooling)	67.56	49.13	66.44	54.54	43.46	59.82
Cycle efficiency (% Carnot cycle) 40-105°F	90.5	83.2	81.8	87.5	84.9	82.0
Solubility of water in refrigerant	Negligible	Negligible	Negligible	Negligible	Negligible	Negligible
Miscibility with oil	Miscible	Miscible	Limited	Miscible	Miscible	Miscible
Toxic concentration (% by vol.)	Above 10%	Above 20%	*	*	Above 20%	Above 20%
Odor	Ethereal, odorless when mixed with air	Same	Same	Same	Same	Same
Warning properties	None	None	None	None	None	None
Explosive range (% by vol.)	None	None	None	None	None	None
Safety group, U.L.	5	6	5A	4.5	6	5A
Safety group, ASA B9.1	1	1	1	1	1	1
Toxic decomposition products	Yes	Yes	Yes	Yes	Yes	Yes
Viscosity (centipoises) Saturated liquid 95°F	0.3893	0.2463	0.2253	0.5845	0.3420	0.2150
105°F	0.3723	0.2395	0.2207	0.5472	0.3272	0.2100
Vapor at 1 atm 30°F	0.0101	0.0118	0.0120	0.0097	0.0108	*
40°F	0.0103	0.0119	0.0122	0.0098	0.0109	*
50°F	0.0105	0.0121	0.0124	0.0100	0.0111	*
Thermal conductivity, $k^{(1)}$ Saturated liquid @ 95°	0.0596	0.481	0.573	0.512	0.0435	*
105°F	0.0581	0.469	0.553	0.0500	0.0421	*
Vapor at 1 atm 30°F	0.0045	0.0047	0.0060	0.0037	0.0056	*
40°F	0.0046	0.0049	0.0061	0.0039	0.0057	*
50°F	0.0046	0.0051	0.0063	0.0040	0.0059	*
Liquid circulated, 40-105°F (lb/min/ton)	2.96	4.07	3.02	3.66	4.62	3.35
Theoretical displacement, 40-105°F (cu ft/min/ton)	16.1	3.14	1.98	39.5	9.16	2.69
Theoretical horsepower per ton, 40-105°F	0.676	0.736	0.75	0.70	0.722	0.747
Coefficient of performance 40-105°F (4.71/hp per ton)	6.95	6.39	6.29	6.74	6.52	6.31
Cost compared with R-11	1.00	1.57	2.77	2.15	2.97	2.00

* Data not available or not applicable

(1) k in $\text{Btu}\cdot\text{ft}/\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F}$

NOTES:

17

FRACTIONATION AND ABSORPTION FUNDAMENTALS

The detailed design of fractionators and absorbers is a specialty area that involves details outside the scope of this book. Many books and endless technical papers discuss the details. This chapter is an overview of the basic factors governing performance for those who specify, buy and operate such units.

An absorber is any device where a liquid (absorbent) flows countercurrent to a gas stream for the purpose of removing one or more constituents (absorbate) from that gas. Most absorbers are vertical with the liquid entering at the top and the gas at the bottom. The amount of contact that must be provided in this absorber tower depends on the system, the relative flow rate of liquid and gas, and the concentrations involved.

A fractionator is a device for splitting a feed into two or more parts, at least one of which will have a controlled composition or vapor pressure. In crude oil or condensate systems a fractionator that is called a *stabilizer* is an alternative to stage separation. The fractionator is essentially a constant pressure device that uses temperature and the required vapor-liquid contact to produce the desired feed split.

Both are mass transfer type devices that must operate according to the rate criteria outlined in Chapter 12.

$$Q_m = k_G A (c_1 - c_2) \quad (17.1)$$

Where: Q_m = mass transfer rate between phases
 k_G = mass transfer coefficient (empirical)
 A = area of contact between vapor and liquid
 c_1 = concentration (high)
 c_2 = concentration (low)

(In using Equation 17.1 one never truly knows a value of "A." Consequently, the mass transfer coefficient often is written k_{GA} , found by dividing measured mass rate by the concentration difference.)

Mass always flows from high concentration to low concentration. In an absorber the concentration of the absorbate must be higher in the vapor than in the liquid for absorption to occur. In *stripping*, the opposite of absorption, the liquid concentration must be the highest of the two. Concentration is the *driving force*. The term "concentration" as used in this paragraph means concentration relative to equilibrium.

The resistance to transfer is the reciprocal of area. The empirical mass transfer coefficient includes the effect of all variables except concentration and area. For a given system operating at specified conditions with required characteristics of the effluent streams, the efficiency of mass transfer depends on area of contact between vapor and liquid. Control of this area, the temperature gradient and vapor-liquid ratios, is the major suite of variables available to the designer.

For a fractionator with a given feed composition and condition, at a fixed pressure, the basic variables available to the designer are control of the temperatures at top and bottom, feed location and the amount of vapor-liquid contact. In an absorber the basic variables are absorbent rate and composition, and the amount of vapor-liquid contact.

There are three basic ways to maximize the amount (area) of vapor-liquid contact per unit volume of tower:

1. Bubble the gas through liquid.
2. Spray the liquid as droplets into the gas stream.
3. Flow the liquid as a thin film over a surface with a high area.

Many different mechanical configurations have been developed to accomplish these. Competitors continually search for new ways to obtain advantage by increasing contact efficiency. In fractionation and absorption, (1) and (3) are the primary methods employed.

TRAY (PLATE) TYPE TOWERS

This is a vertical vessel containing a series of trays. Liquid on the top tray is provided by the absorbent or from partial or total condensing of the effluent gas. The liquid flows across the tray and then by gravity to the next tray down, using what is called a *downcomer*. It flows across the second tray and repeats the same path until emerging from the bottom. On each tray a mechanism is provided for the upward flowing gas to bubble through the liquid crossing the tray.

The upper portion of Figure 17.1 shows the basic pattern, using two different types of downcomers. Weirs are used to help maintain the proper level of liquid on the trays.

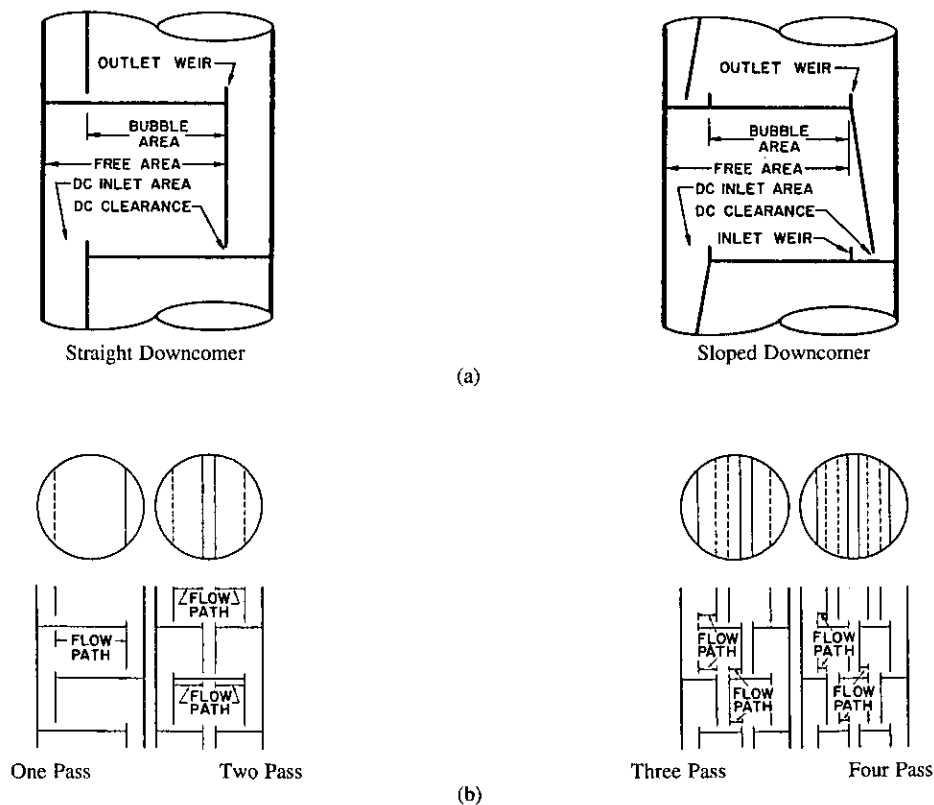
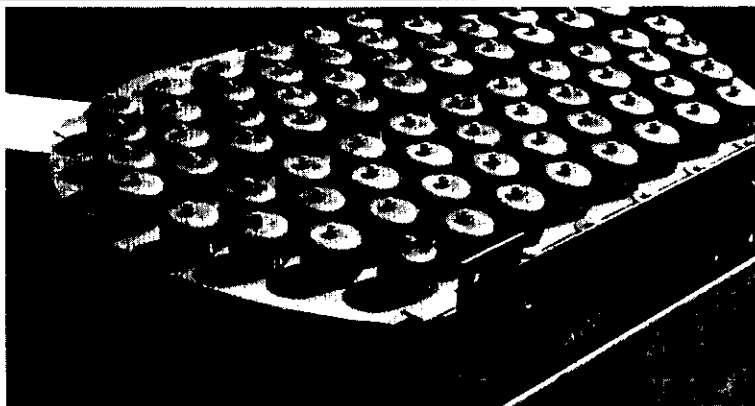


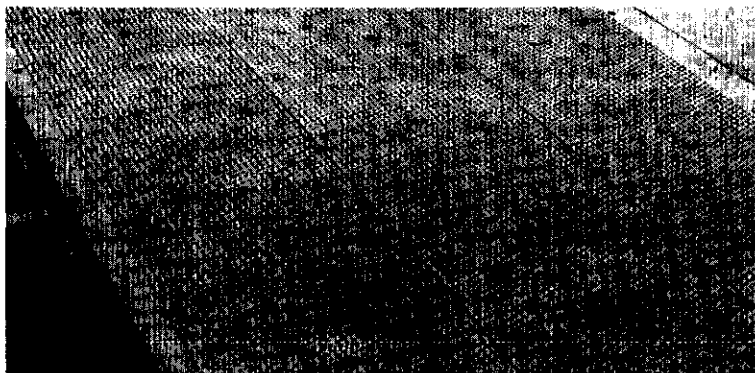
Figure 17.1 Basic Layout of Tray Type Towers

As the contactor diameter increases and/or high liquid rates are involved, it may be impractical to flow the liquid across the total tower because of the pressure drop involved. Therefore, the tray is split so that more than one pass is used, as shown in the lower part of Figure 17.1. This shortens the flow path and reduces pressure drop per tray. But, it reduces the area available for gas flow. Proper design involves balancing many factors like these.

Figure 17.2 shows three types of contact trays used. *Bubble caps* have a riser in the middle of the cap through which the gas passes. This gas hits the top and flows out through the slots. The riser keeps the liquid from flowing through the gas opening, which is called *weeping*. As the name implies, *sieve trays* are nothing but a series of holes drilled in flat metal. Gas bubbles through the holes. The size, number and spacing of holes must be proper to obtain good gas-liquid contact and prevent weeping.



Bubble Cap



Sieve



Ballast (Valve)

Courtesy Glitch, Inc.

Figure 17.2 Three Common Types of Contact Trays

The Ballast tray shown is one manufacturer's version of a *valve tray*, the generic description where the contact device moves with gas rate. There are many types and configuration which are proprietary to a given manufacturer. The "valve" rests on the tray deck in the closed position until gas impact is sufficient to cause it to open. It is, in effect, a variable orifice whose opening depends on gas rate up to the limit imposed by the mechanism which holds it in place.

Both sieve and valve trays are less expensive than and can be more efficient than bubble caps, but weeping can present a problem at low gas or liquid rates. Factors governing choice and design will be discussed in later sections of this chapter.

PACKED TOWERS

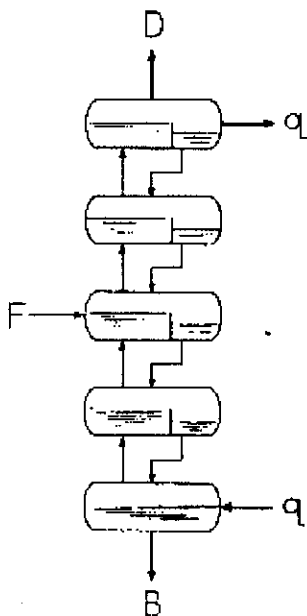
These are basic alternatives to tray-type units. The packing may be *structured* or *random*. Whatever the type, said packing must be capable of producing a thin liquid film and a long, tortuous flow path even at low flow rates. It also should exhibit a low pressure drop and liquid holdup. Structurally, it should not degrade thermally, be chemically impervious to the fluids involved and not crush or powder mechanically. There has been a substantial evolution in packing in recent years.

The right-hand portion of Figure 17.3 shows various common types of random packing. Random packing can be made from plastics like fiberglass reinforced polypropylene and corrosion-resistant metals.

This type consists of individual pieces which are randomly packed into a container. Structured packing comes in units made for placement in the unit. It is made from knitted mesh and all sorts of metal plate and tubing which is crimped, twisted, wound, stamped or somehow arranged to present a large area per unit volume. Each is described by a name which is a registered trademark of its owner. Names like Mellapak, Gempack, Flexipac, Montz, and Flexigrid are just a few of the structured packings available.

The left side of Figure 17.3 shows all of the elements that may exist in a packed tower.^(17.1) Shown are various types of liquid distributors and support plates. Liquid distribution is a problem. The type of distributor chosen and the number used may be critical to obtain satisfactory tower performance.

In some service there is a tendency to retrofit some tray contactors with modern packing. This may expand tower capacity and efficiency. However, packed towers have some inherent limitations. It is more difficult to predict the performance of packed towers. In some applications their flexibility is limited. A later section discusses further packing details.

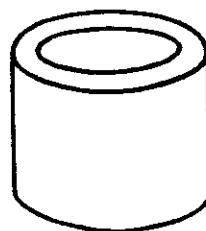
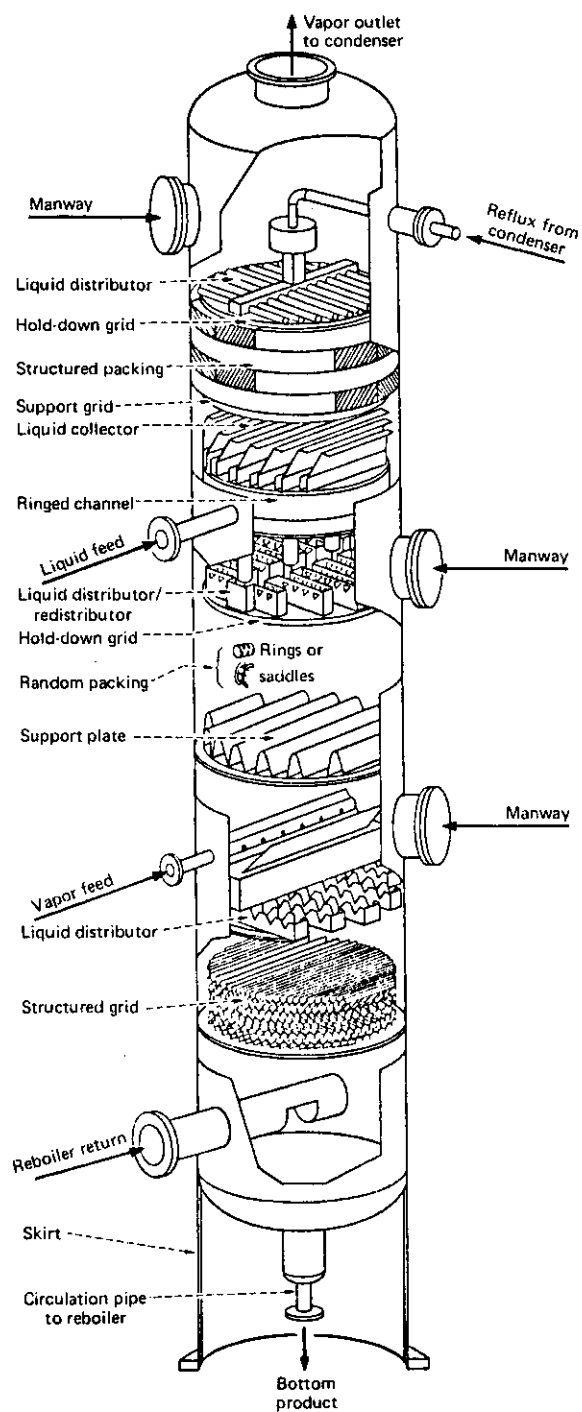


USE OF EQUILIBRIUM CONCEPT

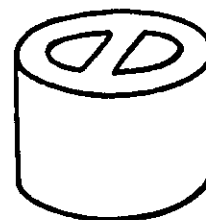
The concentration of vapor and liquid is different at each point in the contactor. It is convenient (and useful) to use the concepts of equilibrium and the energy balance to describe contactor performance.

This is done by assuming that the tower is composed of a series of equilibrium stages that will produce the overall results desired. The stages can be considered a series of "separators." Gas enters at one end and flows across. Gas fed into the bottom contacts the liquid and leaves in equilibrium with the liquid leaving. As shown, the liquid and gas flow countercurrent to each other.

Heat exchange must occur at each stage. Energy is released by each unit mass condensing to liquid; it is absorbed by each unit mass vaporizing. The *heat of sorption* is the latent heat of the components



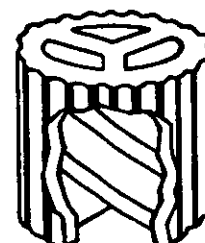
Rasching Ring



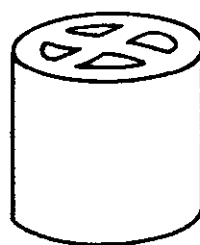
Lessing Ring



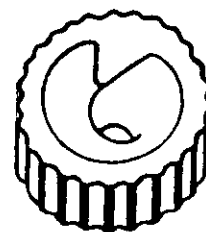
Double Spiral Ring



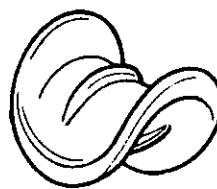
Triple Spiral Ring



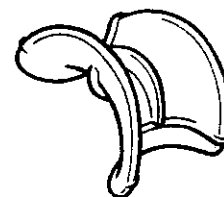
Cross-Partition Ring



Single Spiral Ring



Berl Saddle



Intalox Saddle

Figure 17.3 Towers Using Structured and/or Random Packing

involved, at the average P and T of the stage, plus an extra amount of energy to permit absorption or desorption to occur. The pressure does not vary much between stages but the temperature may do so.

In stage separation of the type discussed in Chapter 5, a series of pressure changes is used. The only change in temperature is that which could occur across the dump valves. This can separate the feed into fluids possessing specified overall physical characteristics but it is impossible to have any degree of control over composition.

The contactor consists of a series of essentially constant pressure "separators." Temperature and the relative gas-liquid rate can be varied to produce a specified composition or vapor pressure material.

For each equilibrium stage the vapor and liquid leaving are related by K values at the P and T of that stage. A mass and energy balance also must be written for each stage. This is a trial-and-error process that ordinarily is performed on a computer. The balance is performed until it "closes," in that one has the required number of equilibrium stages to fit the external conditions imposed for the purpose of making the calculation. The number of such equilibrium stages will depend on:

- Feed composition, phase and flow rate (in fractionator)
- Inlet gas P, T, composition and rate
- Inlet liquid P, T, composition and rate
- Outlet gas specifications
- Outlet liquid specifications

Note that all of the above can be specified. Some variables are fixed by the overall mass and energy balance around the contactor.

It is desirable to calculate the number of theoretical stages for a range of possible conditions. The optimum number is the one which attains the desired goals at minimum cost per unit time or per unit processed.

In a tray type contactor, the term *theoretical tray* (or plate) is used. For a packed contactor the HETP (height equivalent of a theoretical plate) and HTU (height of a transfer unit) is used.

Efficiency

Once one has obtained the number of theoretical stages required, this must be converted to an actual amount of contact (trays or packing). Equilibrium cannot be achieved on an actual tray or its equivalent; the vapor and liquid are not in contact for a long enough time. Thus, some efficiency must be used to relate theoretical to actual.

One approach is to use a Murphree type efficiency which measures a ratio showing the approach to equilibrium on a given tray. It will vary from tray to tray.

Overall efficiency is used most commonly. It is expressed by the equation

$$\text{Overall Efficiency} = \frac{\text{No. of Theoretical Trays}}{\text{No. of Actual Trays}} \quad (17.2)$$

Typical overall efficiency values will be discussed in later sections.

FRACTIONATION

Regardless of how the fluids are removed from natural gas, fractionation is necessary if products that meet any kind of rigid specification are to be made. The number of fractionating columns required depends on the number of products to be made and the character of the liquid which serves as feed. The single tower system shown in Figure 17.4(a) ordinarily produces one specification product from the bottom stream, with all other components in the feed passing overhead. This type of fractionator (stabilizer) is ordinarily used where a natural gasoline or stable liquid is being produced for sale to a pipeline.

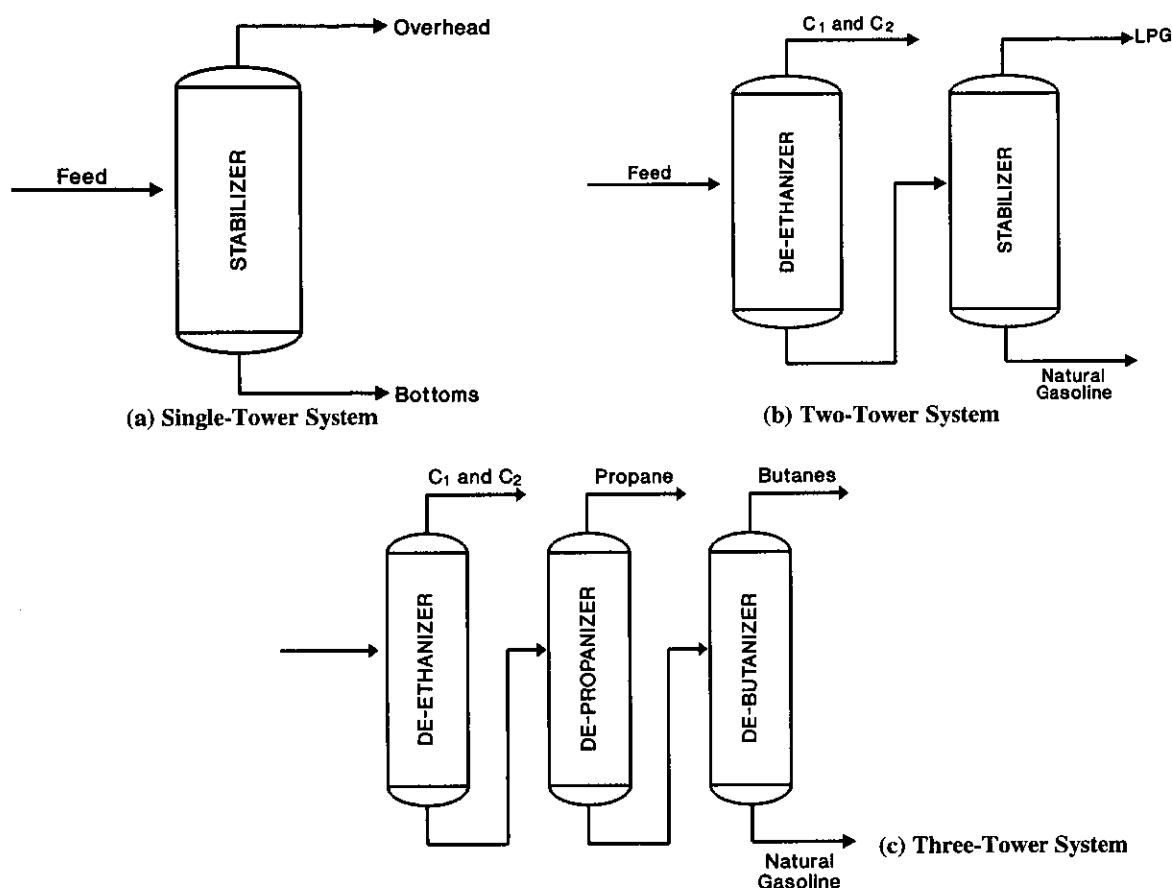


Figure 17.4 Typical Fractionation Systems

The two-tower system shown in Figure 17.4(b) is most commonly used to produce an LPG mixture overhead and a natural gasoline product as the bottoms. In this system, the deethanizer must remove all methane, ethane, and other constituents not salable in the two product streams from the second tower. Any material that enters the second tower must necessarily leave in one of the product streams.

The three-tower system shown in Figure 17.4(c) most commonly produces commercial propane, commercial butane and natural gasoline as products. In this system also, the deethanizer must work properly to remove all constituents that cannot be sold in one of the three products. The sequence of fractionation following the deethanizer may be varied. In the second tower, an LPG mixture could be produced overhead with natural gasoline produced as bottoms. The third tower would then split the LPG into commercial propane overhead and commercial butane as bottoms. This sequence is favored sometimes where the market situation is variable and a market for LPG only exists during a portion of the year. During this period, the third tower would be shut down and not operated.

If ethane product was saleable, the deethanizer would be proceeded by a demethanizer which would remove methane and lighter components from the NGL. Due to the difficulty in condensing reflux for a demethanizer, this column is often a stabilizer with a top tray feed, rather than a conventional fractionator.

Figure 17.5 is a schematic view of a typical fractionator. The bottom product (bottoms) is produced through a reboiler at a temperature which corresponds to the bubblepoint of the bottom product at tower pressure.

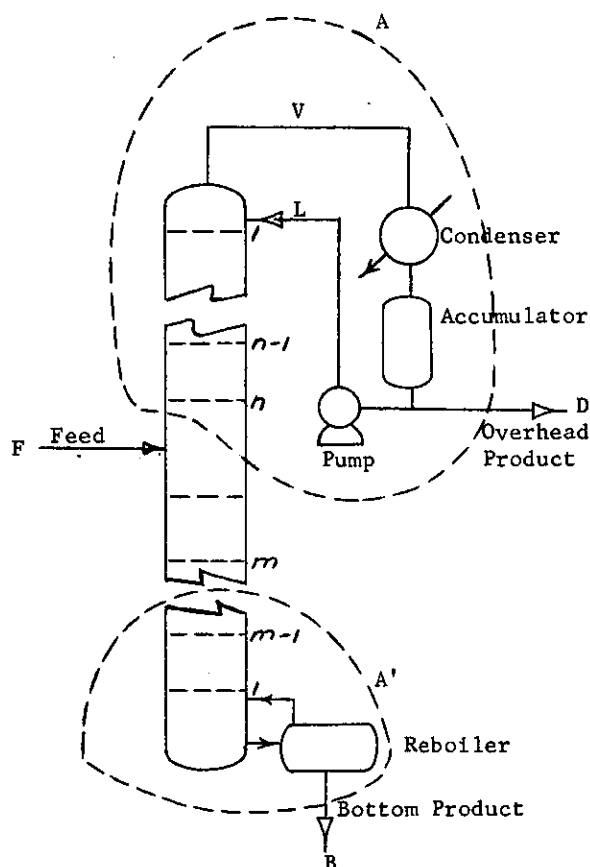


Figure 17.5 Fractionator Nomenclature

The overhead product (distillate) leaves as a saturated liquid (total condenser) or a saturated vapor (partial condenser). A portion of the vapor leaving the top tray of the tower is always condensed and returned to the tower as reflux. With a total condenser, the reflux and the distillate product will have the same composition. In the case of a partial condenser, the reflux will be the liquid in equilibrium with the distillate vapor product at condenser conditions. The reflux furnishes a controlled amount of liquid for the trays in the column above the feed and provides the mechanism by which better separation is made than is the case with a non-refluxed tower. The reboiler provides vapor for the tower which serves the same purpose in the stripping section (below the feed) as the reflux does for the rectifying section (above the feed). The amount of reflux and reboiled vapor and the number of trays must be balanced to achieve the desired terminal products from the tower.

The operating pressure of a fractionation tower is ordinarily fixed by the condensing temperature of the overhead product. Temperature in the condenser is ordinarily controlled by the cooling medium. Allowing for sufficient temperature difference between the cooling medium and the overhead product, condenser temperature is fixed by the designer. In the case of a liquid distillate, the bubblepoint pressure is then calculated; for a vapor distillate product the dewpoint pressure would be calculated. This pressure is the minimum pressure at which the tower can operate at the chosen condenser temperature.

The choice between a total and partial condenser for a tower is a practical matter involving cost and uses for the overhead distillate product. At a given pressure, the dewpoint is always a higher temperature than the bubblepoint, and this tends to minimize cooling costs, all other things being equal.

Design Considerations

The design of a fractionator involves the following discrete steps:

1. Knowledge of the feed composition, quantity, temperature and pressure.
2. Make product "splits" for tower based on feed and desired product needs.
3. Establish a reasonable condensing temperature for the overhead stream.
4. For a liquid distillate product, calculate the bubblepoint pressure at the temperature of Step 3. If the distillate product is a vapor, calculate the dewpoint pressure. After calculating the pressure, adjust to operating conditions. For example, if a pressure of 1.2 MPa were

calculated, this probably would be adjusted upward to 1.3 or 1.4 MPa as an operating pressure for the actual column.

5. Assume the pressure is constant throughout the column and calculate the bubblepoint temperature of the bottoms.

NOTE: At this point, review the assumption of Step 2 and the results of Steps 3, 4, and 5 which depend on it to see that the values are compatible with the rest of the facility. Frequently, Steps 2-5 will have to be repeated to properly balance the plant heating and cooling loads.

6. Calculate the minimum number of trays and the minimum reflux rate needed to produce the desired products.
7. Calculate the combination of actual reflux and actual theoretical plates needed.
8. Make a heat balance around condenser to determine condenser duty and around the tower to determine reboiler duty.
9. Size the column.

This procedure is set up for the operating employee who is responsible for specifying, choosing and operating a fractionation system. Detailed mechanical design by the equipment vendor involves details and specialized knowledge normally beyond the scope of the purchaser. The above procedure and its application, which is the concern of the rest of this chapter, is designed for the operating employee. Featured are a series of "short-cut" methods suitable to one's time availability and needs.

If the tower involved is the first one in a fractionation system, the conditions of the feed to the column will be fixed by the separation process. A surge vessel prior to this tower might change the analysis of the feed if some vapor is withdrawn at that point. (An example of this is a rich oil flash drum situated between the absorber and the deethanizer.) For a system containing several towers, the split desired in each column should be made before completing analysis of any one. This assures that the splits set up for the different towers produce satisfactory products in all streams. A perfect separation between adjacent components cannot be specified. This will lead to a situation impossible to achieve in an actual column. In producing propane, for example, one must allow a small amount of ethane and butanes to be present. In this case, however, the propane must still meet the purity specifications demanded.

In fractionator design, a material balance around the column must be made first. In a multi-component mixture, the two components between which a separation is desired are spoken of as the "key" or "cut" components. The light key component is ordinarily the lightest component (lowest boiling) that appears in appreciable quantity in the bottom stream; the heavy key component is the heaviest (highest boiling) that appears in appreciable quantity in the distillate stream. Normally, these two components are adjacent to each other in the volatility listing of the components in the feed. A complete separation between the key components is impracticable. Experience is the only guide in determining a separation which can be accomplished in a column of reasonable size with reasonable reflux and reboiler duties.

For a fractionator operating at steady state, it can be shown that

$$\frac{D}{F} = \frac{x_F - x_B}{x_D - x_B} \quad \text{and} \quad \frac{B}{F} = \frac{x_F - x_D}{x_B - x_D} \quad (17.3)$$

Where:

- D = distillate rate, mols
- B = bottoms rate, mols
- F = feed rate, mols
- x_D = mol fraction of a component in the distillate
- x_B = mol fraction of a component in the bottoms
- x_F = mol fraction of a component in the feed

The design of a fractionator requires two specifications. For non-refluxed distillation only one specification is required. There are several ways in which the specifications can be stated. Three are listed below.

1. Specified recovery of two components
2. Specified composition of two components
3. Specified recovery and composition of one component

Vapor pressure is sometimes used as a specification in place of composition. Product splits can be made using the procedure outlined in Chapter 5.

In some cases, particularly existing towers, one of the specifications can be reflux rate, reboiler or condenser duty. In other words, an equipment limitation constrain the operation of the tower. When this happens, one (or both) product specifications is no longer relevant.

An example calculation to determine the split for each of these cases will be made to illustrate the way in which the specifications can be converted into product splits necessary for the start of the distillation calculation.

Case I Specified Recovery

Consider the following example for a depropanizer. With the feed shown below, we desire to recover 81% of the propane in the distillate and 99.4% of the isobutane in the bottoms.

For Propane: Mols to Distillate = $(0.81)(8.9) = 7.2$
 Mols to Bottoms = $(0.19)(8.9) = 1.7$

For Isobutane: Mols to Distillate = $(0.006)(63.3) = 0.4$
 Mols to Bottoms = $(0.994)(63.3) = 62.9$

The column material balance will be

Component	Feed mols	Distillate		Bottoms	
		Mols	Mol %	Mols	Mol %
C ₂	0.2	0.2	2.6	0.0	0.0
C ₃	8.9	7.2	92.3	1.7	1.8
iC ₄	63.3	0.4	5.1	62.9	68.3
nC ₄	27.6	0.0	0.0	27.6	29.9
	100.0	7.8	100.0	92.2	100.0

In calculating product compositions, it has been assumed that the non-key components (C₂ & nC₄) do not distribute. In other words, no material lighter than the light key will be present in the bottoms and none heavier than the heavy key in the distillate. This will not be the actual case. There will be trace amounts of lighter components in the bottoms and heavier components in the distillate. As a practical matter, however, the amounts of these materials are so small that they can be neglected safely in most calculations.

Case II Specified Composition

Using the same feed composition, determine the overhead and bottoms compositions when producing a distillate product that contains at least 95% propane, and a bottoms product which contains less than 1% propane.

Equation 17.3 can be used to establish the required distillate to meet the specifications:

$$\frac{D}{F} = \frac{x_F - x_B}{x_D - x_B} = \frac{8.9 - 1.0}{95.0 - 1.0} = 0.084 \quad , \quad \text{distillate rate} = 8.4 \text{ mols}$$

Component	Feed mols	Distillate		Bottoms	
		mols	mol %	mols	mol %
C ₂	0.2	0.20	2.4	0.0	0.0
C ₃	8.9	7.98	95.0	0.92	1.0
iC ₄	63.3	0.22	2.6	63.08	68.9
nC ₄	27.6	0.0	0.0	27.6	30.1
	100.0	8.40	100.0	91.60	100.00

Since C₃ must be at least 95% of the distillate; $d_{C_3} = (0.95)(8.4) = 8.0$ mols

In some cases, the specification may be expressed in terms of two components; e.g. the bottoms product shall contain no more than 1% C₃ and the distillate product no more than 1.5% iC₄. This type of specification also fixes the fractionator for material balance and can be solved by developing two simultaneous equations incorporating the material balance and product specifications.

Case III Specified Recovery and Composition

In the depropanizer, we wish to recover 95% of the propane in a distillate product with a purity of 94%.

$$\text{mols of C}_3 \text{ in distillate} = (0.95)(8.9) = 8.46$$

$$\text{total distillate mols} = 8.46/0.94 = 8.99$$

$$\text{mols iC}_4 \text{ in distillate} = 8.99 - 8.46 - 0.20 = 0.33$$

The column material balance will be

Component	Feed mols	Distillate		Bottoms	
		mols	mol %	mols	mol %
C ₂	0.2	0.20	2.2	0.00	0.0
C ₃	8.9	8.46	94.0	0.44	0.5
iC ₄	63.3	0.33	3.7	62.97	69.2
nC ₄	27.6	0.00	0.0	27.60	30.3
	100.0	8.99	100.0	91.01	100.0

For all three of these examples, propane and isobutane would be chosen as the key components if multicomponent short-cut calculations were to be made.

The column top temperature depends upon the operating pressure of the column, which in turn depends on the cooling medium available for condensing reflux. Most columns are designed to use cooling water or air, so a design temperature between 32-55°C is usually used as the temperature of hydrocarbons leaving the

condenser. A partial condenser offers the advantage of lower column pressure. The reduction in column pressure, however, is appreciable only for a distillate product of wide boiling range. For most towers producing specification products, the difference in the dewpoint and bubblepoint for the overhead stream is small. Column pressure will be the pressure at the outlet of the condenser plus the pressure drop through the condenser and overhead vapor line.

Example 17.1: Determine the operating pressure of the depropanizer for Case I, using a total condenser operating at 54°C.

For a total condenser the accumulator pressure is determined by performing a bubblepoint calculation on the distillate product. (A dewpoint calculation is used for a partial condenser). Assuming a 100 kPa pressure drop between accumulator and column, the column pressure will be 1928 kPa.

Comp.	x_D	K's at 1928 kPa & 54°C	Kx
C ₂	2.6	2.7	7.1
C ₃	92.3	0.98	90.5
iC ₄	5.1	0.47	2.4
	100.0		100.0

The vapor from the top plate has the same composition as the distillate product when a total condenser is used. When a partial condenser is used, the condenser will operate at the dewpoint of the distillate. In either case, the last vapor liquid separation in the tower will be at the dewpoint of the distillate product. This temperature will be 61°C as shown in the dewpoint calculation.

Comp.	y_1	K's at 1928 kPa & 61°C	y/K
C ₂	2.6	2.8	0.9
C ₃	92.3	1.04	89.0
iC ₄	5.1	0.51	10.0
	100.0		99.9

The bottom temperature of the column (reboiler temperature) is obtained by a bubblepoint calculation on the bottom product. For the depropanizer, the bottom temperature will be 105°C as shown from the bubblepoint calculation.

Comp.	x_B	K's at 1928 kPa & 105°C	Kx
C ₃	1.8	1.87	3.4
iC ₄	68.3	1.05	71.4
nC ₄	29.9	0.84	25.2
	100.0		100.0

From a fractionation standpoint, optimum column operation is obtained when the feed enters the column at its bubblepoint. Assuming that the feed to the column will be at its bubblepoint for the depropanizer, this temperature turns out to be 100°C.

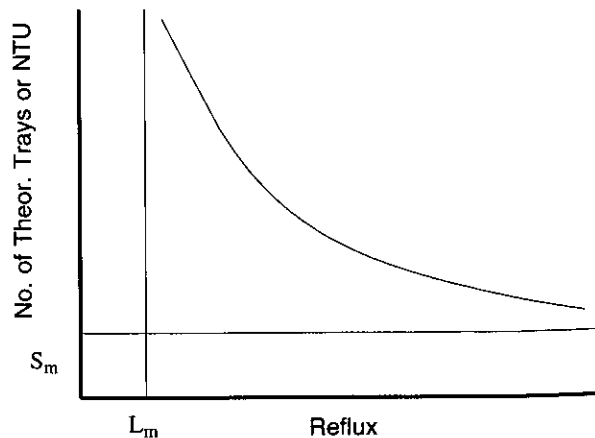
Comp.	x_F	K's at 1928 kPa & 100°C	Kx
C ₂	0.2	4.0	0.8
C ₃	8.9	1.77	15.8
iC ₄	63.3	0.98	61.8
nC ₄	27.6	0.78	21.6
	100.0		100.0

Notice in Example 17.1 that the feed temperature is very nearly equal to the reboiler temperatures. This is because the bottoms product to feed ratio is almost one (≈ 0.92). As a result, one would expect the feed tray on this tower to be near the bottom of the tower.

Theoretical Plates vs. Reflux

Once the foregoing preliminary calculations have been completed, the next step is to determine the combination of number of theoretical plates and reflux necessary to make the desired separation between the key components. There are an infinite number of combinations of these two variables that will give the desired separation, as illustrated in the figure at right.

The curve is essentially a hyperbola. At one asymptote, it approaches the minimum reflux rate and at the other it approaches the minimum number of trays. The minimum reflux rate occurs at that value of reflux with an infinite number of trays in the column to produce the desired separation between the key components. The minimum number of trays occurs at total reflux when the separation per plate in the tower is a maximum. The definition of, and reasons for, these two extremes of column operation will become clearer as the discussion below unfolds.



Neither of these two limits of operation in the column has practical value in itself. However, they do serve as useful correlation tools for predicting the relationship between operating reflux and actual number of theoretical plates. Most hydrocarbon fractionators operate at an actual reflux from 1.05 to 1.25 times the minimum. Operation is toward the minimum reflux rate to minimize heating and cooling costs. Condenser heat load is a direct function of reflux rate; as this increases, the reboiler load also has to increase to maintain the energy balance around the tower.

SHORT-CUT CALCULATIONS

There are some short-cut methods suitable for the calculation of general behavior. Those which follow have been found useful and reliable for the types of fractionation systems discussed in this book. They calculate minimum reflux (L_m), minimum theoretical stages (S_m), the relationship between reflux and contact, and overall efficiency. They calculate a multicomponent system as a pseudo-binary consisting only of the light key and heavy key components.

Appendix 17A reviews the McCabe-Thiele method for binary mixtures, which can be used also for multicomponent mixtures. It is not widely used commercially, but it contributes visually to an understanding of fractionator performance.

Shortcut calculations are useful for:

1. Scoping studies suitable for preliminary costs
2. Parametric evaluation of operating variables
3. Separations having coarse purity requirements (i.e., contaminants > 0.5 mol%)
4. Detailed designs for ideal and close-to-ideal systems
5. Designs for systems for which equilibrium data are unavailable

On the other hand, *rigorous* design procedures should be applied for the following:

1. The separation is a multicomponent one, requiring high product purity.
2. The system is highly nonideal but good equilibrium data are available.
3. The relative volatility between key components is less than 1.3.
4. One or more of the components is near the critical pressure.

Fenske's Method for Minimum Theoretical Plates

The Fenske Equation is a convenient and very useful technique for calculating the minimum number of theoretical plates required for most multicomponent separations. When properly used, it can be applied between the top and bottom of the distillation column and give a very accurate estimate of the minimum number of theoretical plates. With a total condenser, it is applied between the top plate of the tower and the reboiler; for a partial condenser, it is applied between the reboiler and the distillate product. The Fenske Equation can be written in several forms. The most convenient form is:

$$S_m = \frac{\log \left[\left(\frac{x_{LK}}{x_{HK}} \right)_D \left(\frac{x_{HK}}{x_{LK}} \right)_B \right]}{\log \alpha_{avg}} \quad (17.4)$$

Where: S_m = minimum number of theoretical plates
 D = distillate product
 B = bottom product
 x_{LK} = mol fraction of light key component
 x_{HK} = mol fraction of heavy key component
 α_{avg} = relative volatility at average tower temperature
 (top tray temperature + bottom tray temperature)/2

Relative volatility is defined as the K value of the light key divided by the K value of the heavy key at a given condition. It is a measure of ease of separation.

Applying Fenske's Equation across the column for Case I of the example depropanizer for which the splits were calculated earlier in the chapter, we have:

$$\begin{aligned} T_1 &= 61^\circ\text{C} \text{ (Dewpoint of distillate)} \\ T_2 &= 105^\circ\text{C} \text{ (Bubblepoint of bottoms)} \\ T_{avg} &= (T_1 + T_2)/2 \\ &\quad @ 83^\circ\text{F and 1928 kPa, } K_{C_3} = 1.44 \text{ and } K_{iC_4} = 0.76 \\ T_{avg} &= (61 + 105)/2 = 83^\circ\text{C} \\ \alpha_{avg} &= 1.44/0.76 = 1.89 \\ S_m &= \frac{\log \left[\left(\frac{92.3}{5.1} \right) \left(\frac{68.3}{1.8} \right) \right]}{\log 1.89} = \frac{2.836}{0.276} = 10.3 \end{aligned}$$

The minimum number of theoretical trays for the separation is 10.3. This represents the total number of vapor-liquid contacts including reboiler and partial condenser (if any), at total reflux.

Underwood Methods for Minimum Reflux

An infinite number of theoretical plates is required at minimum reflux. This means that a large number of these plates are performing negligible separation of components between vapor and liquid. The area of intersection of the operating lines with the equilibrium curve is termed the *pinch zone* or zone of infinitude. Characteristics of these zones are such that there is no change of composition from tray to tray and, as a result, no change of temperature from tray to tray. Also, there is constant molal overflow. All of these characteristics of the rectifying and stripping section pinch zones are of importance in mathematical developments of minimum reflux computation techniques. There are tray-by-tray methods available but these are suitable only for a computer.^(17/13.2) Understandably, these are long and tedious and involve many repetitive computations. Consequently, short-cut calculations are recommended for estimating minimum reflux rates.

The second Underwood equation involves use of two equations.^(17.3)

$$\sum_{i=1}^{i=c} \frac{\alpha_i f_i}{\alpha_i - \phi} = F(1 - q) \quad (17.5)$$

$$\sum_{i=1}^{i=c} \frac{\alpha_i d_i}{\alpha_i - \phi} = L_m + D \quad (17.6)$$

Where: α_i = relative volatility of component at average tower temperature
 ϕ = constant
 f_i = mols of component in feed
 d_i = mols of component in distillate
 q = total heat needed to convert one mole of feed into a saturated vapor divided by the molal latent heat of the feed
 for B.P. feed, $q = 1.0$
 for D.P. feed, $q = 0$
 for 2 phase feed, $0 < q < 1.0$
 L_m = minimum reflux rate, mols
 D = distillate rate, mols

Use of Equations 13.5 and 13.6 to calculate the minimum reflux rate involves determination of the value of ϕ that will satisfy Equation 17.5. This is a trial-and-error solution, as shown in the sample calculation. The procedure is simplified considerably by the limitation that the value of ϕ sought falls between the relative volatilities of the key components ($\alpha_{HK} < \phi < \alpha_{LK}$).

For a bubblepoint feed, Equation 17.5 becomes

$$\sum_{i=1}^{i=c} \frac{\alpha_i f_i}{\alpha_i - \phi} = 0 \quad (17.7)$$

Since the relative volatilities for all components are based on the heavy key, this means that ($1.0 < \phi < \alpha_{LK}$). Thus, the value of ϕ that satisfies Equation 17.5 is used in Equation 17.6 to calculate the minimum reflux. In this calculation, the distillate composition is obtained from the splits that were made for the column. In the case of the example on page 288, the distillate composition was obtained from the split predicted for Case I of the depropanizer example.

The calculated value of 79.8 mols of minimum reflux for 100 mols of feed is rather high. This is caused by the high percentage of recovery (99.4%) of the isobutane in the bottom stream.

The first method of Underwood^(17.4) is summarized in Appendix 17B. It gives values of minimum reflux that consistently are high by 10-20%, and approximate an actual reflux rate. It is easy to do manually and may prove useful for operational calculations.

This calculation of minimum reflux is a very practical, convenient way to begin the analysis of tower performance. It provides one with a reference point for the adjustment of reflux rate. In operational troubleshooting, it provides a clue about whether reflux rate is a potential problem.

Example 17.2: Calculation of ϕ , Assume $\phi = 1.735$

Comp.	Feed mols	K @ 83°C 1928 kPa	α_i	$\alpha_i f_i$	$\alpha_i - \phi$	$\frac{\alpha_i f_i}{\alpha_i - \phi}$
C ₂	0.2	3.48	4.58	0.92	2.845	0.32
C ₃	8.9	1.44	1.89	16.8	0.155	108.52
iC ₄	63.3	0.76	1.0	63.3	-0.735	-86.12
nC ₄	27.6	0.59	0.78	21.5	-0.955	-22.54
	100.0					0.18

Calculation of L_m

Comp.	Distillate Mols	α_i	$\alpha_i d_i$	$\alpha_i - \phi$	$\frac{\alpha_i d_i}{\alpha_i - \phi}$
C ₂	0.2	4.58	0.92	2.845	0.3
C ₃	7.2	1.89	13.14	0.155	87.8
iC ₄	0.4	1.0	0.4	-0.735	-0.5
nC ₄	0.0				0.0
	7.8				87.6

$$87.6 = L_m + 7.8$$

$$L_m = 79.8 \text{ mols/100 mols feed}$$

$$\text{Minimum reflux ratio} = (L_m/D)_m = 79.8/7.8 = 10.23$$

Actual Reflux Rate

Two useful correlations are available for relating actual reflux rate and number of theoretical trays from the corresponding minimum values.^(17.5,17.6)

The Gilliland correlation is widely used. It was originally presented graphically, as shown in Figure 17.6. Once the minimum theoretical stages and minimum reflux have been calculated, the Gilliland correlation can be used to establish an actual reflux rate for a fixed number of stages or conversely, the number of stages for a fixed reflux rate.

An approximate correlation of Gilliland for tray-type towers may be represented by the equation^(17.7)

$$Y = 0.75 (1 - X^{0.5668}) \quad (17.8)$$

Where: $Y = (S - S_m)/(S + 1)$
 $X = (R - R_m)/(R + 1)$
 R = actual L/D
 R_m = minimum L/D
 S = actual theoretical trays
 S_m = minimum theoretical trays

The calculation for packed towers is somewhat different. The following equation may be used^(17.10,11)

$$Y = 0.763 (1 - X^{0.5806}) \quad (17.9)$$

Where: $Y = (NTU - NTU_m)/(NTU + 2A)$
 $A = \ln \alpha/(\alpha - 1)$
 α = relative volatility
 $X = (R - R_m)/(R + 1)$
 NTU = number of transfer units

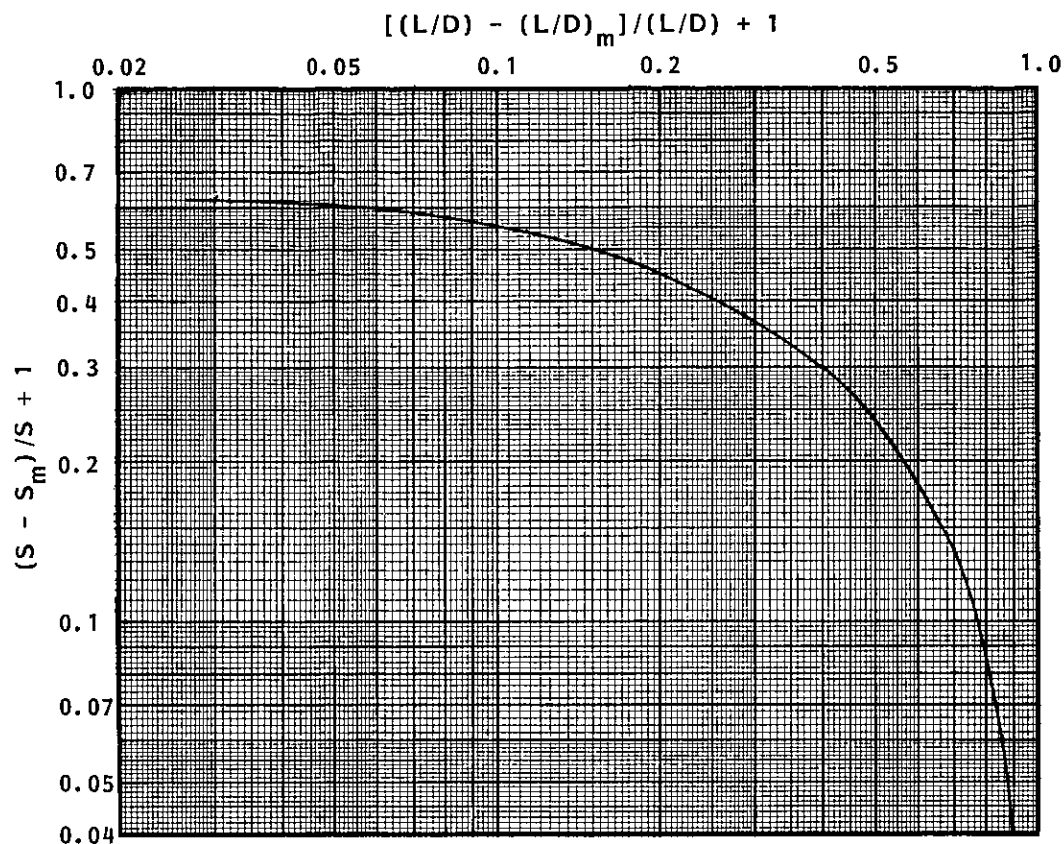


Figure 17.6 Gilliland Correlation for Relationship Between Reflux Rate and Number of Theoretical Plates.

Example 17.3: If the depropanizer in Case 1 is designed to operate at an actual reflux rate 10% above the minimum reflux, how many equilibrium stages will be required to make the separation.

$$(L/D) = 1.1 (L/D)_m = 11.25$$

$$\frac{L/D - (L/D)_m}{(L/D) + 1} = \frac{11.25 - 10.23}{11.25 + 1} = 0.083 \quad , \quad \text{From Figure 17.6}$$

$$(S - S_m)/(S + 1) = 0.57 \quad , \quad S_m = 10.3$$

$$S = 25.3 \text{ equilibrium stages}$$

The reboiler is often equivalent to one equilibrium stage, so for this tower 25.3 equilibrium stages must be installed in the tower itself.

Overall Tray Efficiency

The best source of such efficiency numbers are the results of tests on comparable contactors to the one being calculated. Typically, absorbers have an efficiency of 25-40% and fractionators 50-80%. Said efficiency depends on many factors and may be outside this range. The efficiency of a stripper for deoxygenation of sea water may be as low as 6-8%.

For most common hydrocarbon separations the correlation of O'Connell, shown in Figure 17.7, usually is a good approximation of the efficiency to be expected.^(17.7)

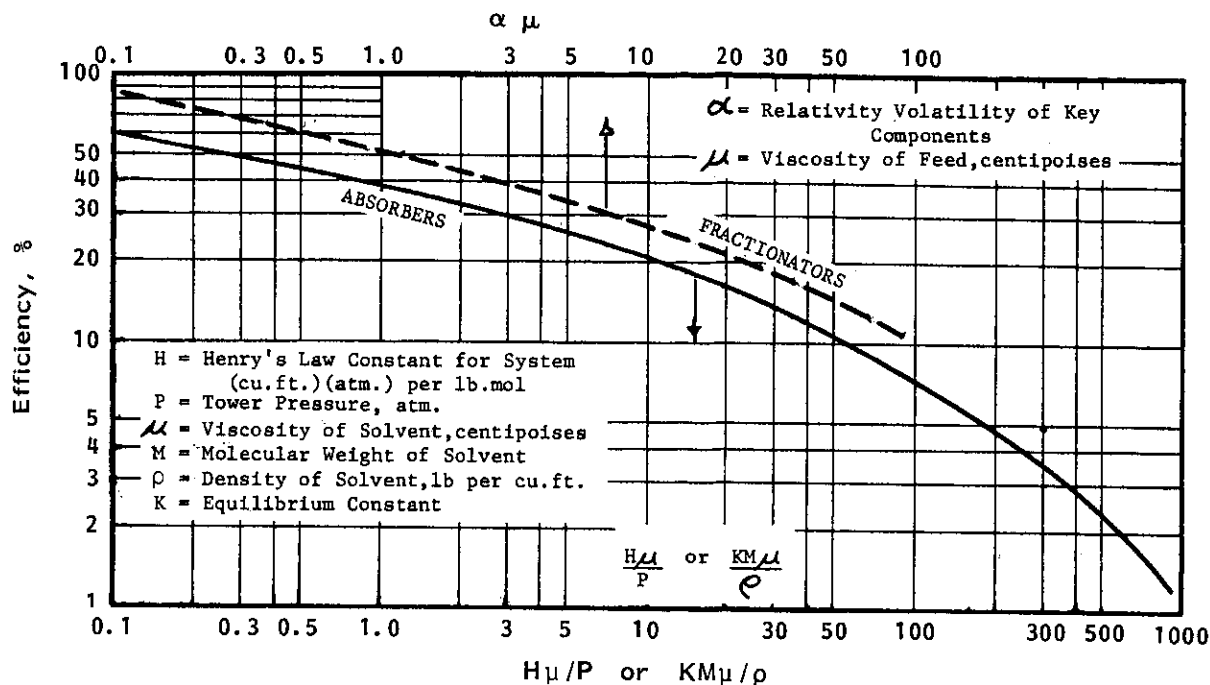


Figure 17.7 Correlation for Overall Efficiency of Absorbers and Fractionators

Location of Feed Tray

The feed tray location should be determined so that feed enters at a temperature so that no sharp change occurs in the column temperature gradient at the feed tray.

There is no exact way to locate a feed plate. There are several calculation techniques which estimate feed location. A convenient empirical correlation is^(17.10)

$$\log \left(\frac{N}{M} \right) = 0.206 \log \left[\left(\frac{B x_{HK_F}}{D x_{LK_F}} \right) \left(\frac{x_{LK_B}}{x_{HK_D}} \right)^2 \right] \quad (17.10)$$

- Where:
- N = no. theoretical stages in rectifying section
 - M = no. theoretical stages in stripping section
 - B = bottoms rate, mols
 - D = distillate rate, mols
 - x_{HK_F} = composition of heavy key in the feed
 - x_{LK_F} = composition of light key in the feed
 - x_{LK_B} = composition of light key in the bottoms
 - x_{HK_D} = composition of heavy key in the distillate

For Case I of the example depropanizer:

$$\log \left(\frac{N}{M} \right) = 0.206 \log \left[\left(\frac{(92.2)(0.633)}{(7.8)(0.089)} \right) \left(\frac{0.018}{0.051} \right)^2 \right]$$

$$\log \left(\frac{N}{M} \right) = 0.206 \log [(84)(0.125)] = 0.21$$

$$N/M = 1.62$$

Approximately 9.7 equilibrium stages below the feed tray (including reboiler) and 15.6 equilibrium stages above.

This relatively high ratio of plates in the rectifying section to plates in the stripping section is caused by the feed composition. The high concentration of heavy key component in the feed indicates that the most difficult part of the separation will be the fractionation of the heavy key component from the distillate product.

Overall Energy Balance

The last step in the basic calculation scheme of the process design for a distillation column is to make an overall heat balance around the fractionator. This is a very important step in determining the desirability and economic feasibility of some of the arbitrary assumptions that were made in the design of the tower. The overall heat balance also has value in evaluating or troubleshooting existing fractionators. The overall energy balance may point out a control problem that exists or some other reason that a tower is operating improperly.

The overall heat balance around the fractionator is

$$Q_B + Q_c = h_D D + h_B B - h_F F \quad (17.11)$$

Where: Q_B = reboiler heat load
 Q_c = condenser heat load
 h_D = enthalpy of distillate product
 h_B = enthalpy of bottoms
 h_F = enthalpy of feed

D, B, F = rate of flow of distillate, bottom and feed streams, respectively

This equation has two unknowns, Q_c and Q_B . All values of "h" are in energy units/mol; "Q" is in energy units/hours. D, B and F are in mols/hour.

An energy balance around the total condenser may be written

$$Q_c = V_1 (h_D - h_1) \quad (17.12)$$

For a partial condenser

$$Q_c = L_0 (h_L - h_1) + D (h_D - h_1) \quad (17.13)$$

Where: h_L = enthalpy of reflux stream
 h_1 = enthalpy of vapor from top tray

The value of Q_c is found from Equation 17.12 or 17.13 and then substituted into Equation 17.11 to solve for Q_B . At this point, the results of all calculations in the process design of the fractionator should be carefully reviewed and studied.

There are many implications of these equations. If one changes the condenser load, reboiler load must change accordingly. The energy balance will apply around the column whether one likes it or not.

Example 17.4: For the depropanizer under Case I conditions:

$$h_1 = 845 \text{ kJ/kg}$$

$$h_F = 675 \text{ kJ/kg}$$

$$h_L = h_D = 575 \text{ kJ/kg}$$

$$h_B = 689 \text{ kJ/kg}$$

For 100 k mols/h of feed, condenser and reboiler duties are

$$\begin{aligned} \text{Condenser} \quad Q_c &= V_1 (h_1 - h_D) = (7.8 + 87.8)(575 - 845)(44.4) \\ &= -1.15 \times 10^6 \text{ kJ/h} \\ &= \underline{-318 \text{ kW}} \end{aligned}$$

$$\begin{aligned} \text{Reboiler} \quad Q_B &= h_D D + h_B B - Q_c - h_F F \\ &= (575)(7.8)(44.4) + (689)(92.2)(57.8) + (1.15 \times 10^6) - (675)(100)(56.8) \\ &= 1.19 \times 10^6 \text{ kJ/h} = \underline{330 \text{ kW}} \end{aligned}$$

FRACTIONATOR CONTROL

The fractionator operates by using a controlled temperature gradient from top to bottom. The composition of the top product is fixed by its bubblepoint or dewpoint. The bottom product is controlled by its bubblepoint.

The fractionator always must operate so that the material and energy balances around it are satisfied on a steady-state basis. Any momentary upsets will be reflected by internal unstable operation which causes intolerable "upsets." Furthermore, it is a "sluggish" device. Liquid "hold-up time" is fairly large since flow rates are relatively low compared to its capacity. Therefore, an inherent time lag occurs when controlling at the tower extremities.

The fractionator must process that feed inlet rate, condition and composition that comes to it. Attempts will be made to control these but the success for this will vary. The fractionator control system must, therefore, be fairly flexible.

A demethanizer and/or deethanizer normally is used to remove noncondensibles that are prohibited in the salable products. The problem is to keep these noncondensibles from passing out the bottom (fairly easy) with only minimum loss of salable products out the top (more difficult). Therefore, the overhead product is the more critical of the two, although both are important.

Usually, a depropanizer and/or debutanizer is producing a commercial overhead product that must meet certain specifications. At this point (hopefully) no noncondensibles are in the system. In the usual situation the propane, butane or LPG mix are less valuable per unit volume than the heaviest product (natural gasoline, condensate, etc.). This latter product should contain all of the propane and butane that the sales specifications allow. Still...the most sensitive control problem is to keep the methane and ethane levels low (from previous fractionation) to meet vapor pressure specs and maintain the heavier ends at a concentration to satisfy weathering tests.

There are several rules that should be followed in fractionator control:

1. The lesser of the two streams should be manipulated (controlwise) to obtain the greatest sensitivity in product quality.
2. Separation should be manipulated to control the purity of the purest product; the material balance should be manipulated to control the quality of the less pure product.

Older control systems attempt to accomplish these functions by the use of pressure, temperature, level and flow controls on each stream independently. The next plateau was to recognize that these streams were

not really independent and to address the interaction between them by means of control loops. The next level of sophistication is to add a chromatograph to sense directly those composition changes that are critical and transmit the proper signal to the controls. A simple analog system may be used to accomplish this. The final plateau is reached by "marrying" all of these to a computer which has been properly programmed. All streams being sensed feed their information into this computer or programmable logic controller which runs through a dynamic simulation and then tells the controls what to do.

A computer does not solve the control problem. It can only react within the limits imposed on it by its creator. It also represents an expensive control system. Many installations simply cannot justify the cost. In others, this degree of sophistication simply is not needed. The difficult decision is finding that proper system for optimum profit.

As a guide in this endeavor, a series of control systems will be shown. These systems should be viewed as examples to illustrate the principles involved, not as firm recommendations.

Feed Surge Control

Regardless of the process used to recover liquid, both flow rate and composition will vary to the first fractionator. A combination surge drum and vessel to flash-off a portion of the methane and ethane might be used ahead of this fractionator. The level must fluctuate in this vessel. One simple approach is to use a liquid level control with a long displacement type float. By setting it on 100-200% proportional control, large level fluctuations will dampen out rate changes.

Diagrams (1) to (4) show several possible arrangements. The pump would be eliminated in the first three if the tank is at a high enough pressure. If the tank is large, (1) could be used. A level indicator with level alarms would be required on the tank to guard against low and high levels. Method (2) shown would use the wide band (and maybe long float) with or without the pump. Method (3) is a further addition that might be necessary when the pressure on the accumulator is not constant. The LC resets the flow recording controller (FRC). In each of these systems, any pump used would have to be of the centrifugal type subject to back pressure control.

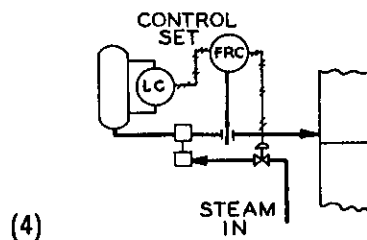
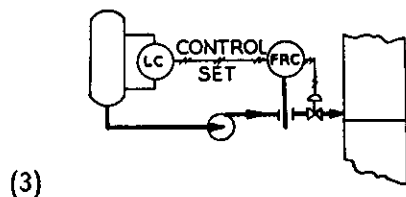
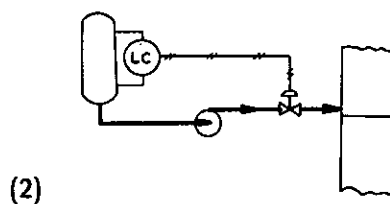
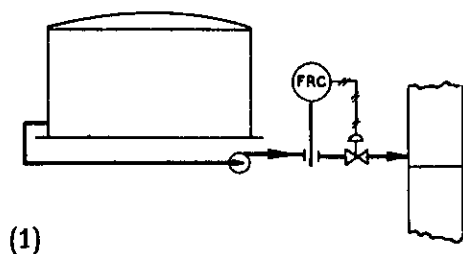


Diagram (4) shows a steam pump layout. The level controller sets the control point for flow. The flow controller actuates the valve on the steam line. Few steam pumps are now used, but this diagram illustrates a simple interlock system.

Column Pressure Control

Regardless of the column control system, it must contain some provision for pressure control. Column pressure can be controlled by manipulating the material balance (rate of distillate product) or by manipulating the condensing temperature (bubble/dewpoint pressure of distillate). Diagrams 5-8 show several arrangements.

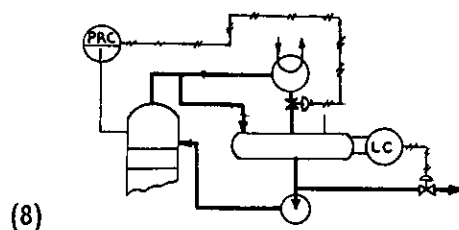
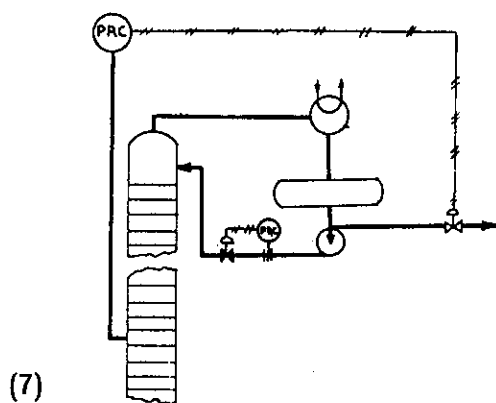
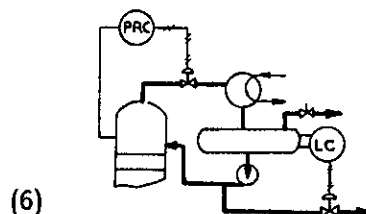
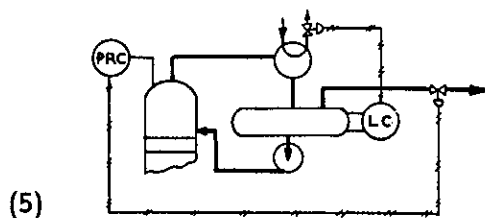


Diagram (5) shows a simple back pressure control on the vapor from the partial condenser. In this case, only enough liquid is condensed to provide reflux. The pressure tap could be on the tower, as shown, or on the reflux accumulator. A proportional response plus reset might be used, although a narrow band proportional control alone could be adequate since pressure offsets are often not critical.

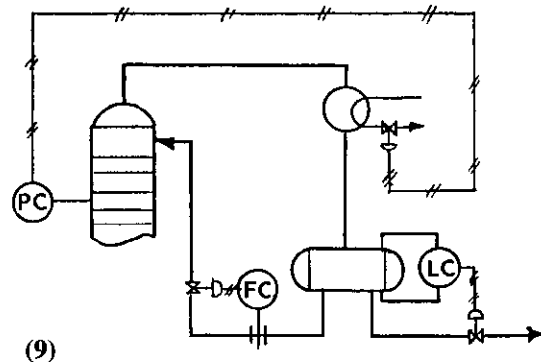
Diagram (6) shows a system for a total condenser that has proven suitable for a narrow boiling range product. The disadvantage is that a large control valve must be placed in the overhead line.

Diagram (7) shows a flooded condenser system for a total condenser. In this system the accumulator runs completely full of liquid and pressure is controlled by manipulating the heat transfer area in the condenser. This method is commonly used in NGL fractionators.

Diagram (8) is an example of one type of hot vapor bypass to control tower pressure. The condenser is partially flooded. The vapor bypass changes the surface temperature of the liquid in the accumulator, hence controlling tower pressure.

The temperature of the condensed product in the accumulator can also be controlled by controlling the medium. This is shown schematically in Diagram (9). This method is not recommended if the cooling medium is cooling water as it induces fouling and scaling in the condenser. If the cooling medium is air, louvers or variable pitch for blades can be used to control air flow. Induced draft coolers are preferred because the tube bundle is not exposed to precipitation.

Most pressure control systems are based on manipulating the cooling rate at the condenser. If the condenser is allowed to operate without restriction, the column pressure will be as low as possible given the cooling medium and operating conditions. This is called "floating pressure control" and has the benefit of reducing the difficulty of separation, as relative volatilities tend to increase with decreasing pressure for most hydrocarbon separations. Reference 17.11 quotes energy savings of 1% for every 1°F reduction in condenser temperature for a separation of iC_5/nC_5 . Reference 17.12 presents one scheme for a floating pressure control system.



With a total condenser, the rate of reflux would be controlled by a flow control set manually or representing a ratio to some stream. On some deethanizers (using a partial condenser), the system might involve setting a reflux rate from the feed rate to the tower.

Column Control

There are a number of ways to instrument a fractionation tower, but the ultimate objectives are

1. Meet the product specifications for the distillate and bottoms products
2. Optimize profitability by minimizing operating costs and maximizing the yield of the most profitable product

In addition the above objectives must be satisfied with a viable, cost effective and safe control system.

To the casual observer, fractionation control appears quite complex but if we ignore feed rate and feed condition, there are only five "knobs to twist" on a fractionation column.

1. Distillate rate
2. Bottoms rate
3. Heat input (reboiler)
4. Heat removal (condenser)
5. Reflux rate

With these five knobs we must control:

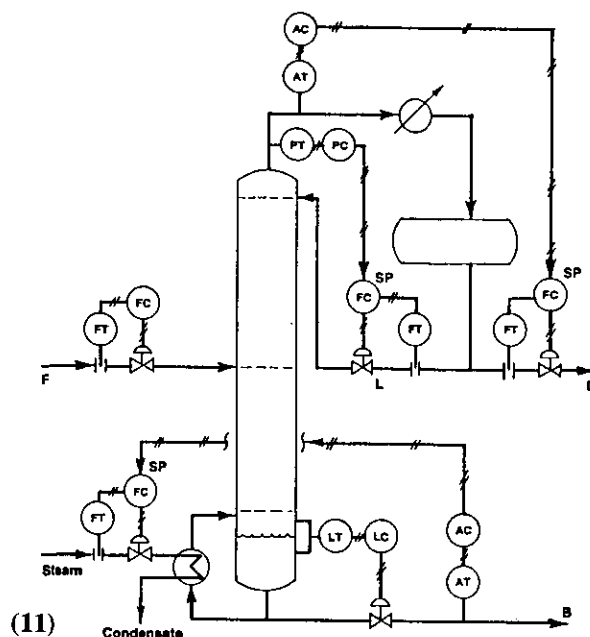
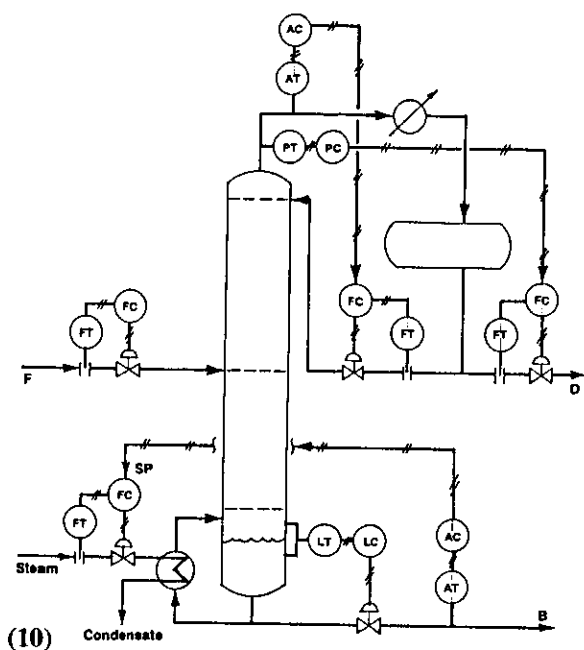
1. Bottoms composition (temperature)
2. Distillate composition (temperature)
3. Tower pressure
4. Liquid level in tower
5. Liquid level in accumulator

Theoretically, there are 120 ways of pairing controlled and manipulated variables to achieve column control. Obviously, many of these are not feasible but several are viable control alternatives.

Historically, the control system has been set up as follows.

Controlled Variable	Manipulated Variable
Bottoms composition (temperature)	Heat input (reboiler)
Distillate composition (temperature)	Reflux rate
Tower Pressure	Heat removal (condenser)
Liquid level in tower	Bottoms rate
Liquid level in accumulator	Distillate rate

If a flooded condenser is used, the liquid level in the accumulator is not controlled and the distillate rate is manipulated to control heat removal (condenser surface area), hence tower pressure. This is illustrated in diagram (10).^(17.13)



Ideally one would like to operate the fractionator so that both the distillate and bottoms product specification are met exactly. We know from our steady-state calculations that this condition allows no degrees of freedom in the fractionator, i.e. the distillate rate, bottoms rate, reboiler duty, reflux rate and condenser duty are all fixed. How does one cope with the upsets – changes in feed rate, composition, ambient temperature, etc. Frequently, the operator will open the control loop on either/both the distillate or bottom composition and make better than specification product. This allows the operating flexibility to handle upsets as well as a cushion of "better than specification" product in the storage tank. This option is certainly viable from a control standpoint but can be quite expensive in terms of energy utilization.

What is preferred is a control system that minimizes energy utilization while meeting product/yield specifications. The system in diagram (10) works well at low reflux rates, but it is highly interactive. The material balance and heat balance interact because distillate flow is used to control the heat removal rate. However, at low reflux rates ($L/D < 1.0$) this system has proven satisfactory.

At higher reflux ratios ($L/D > 5.0$), the conventional system may be inadequate. A material balance system has been used to reduce interaction, as shown in Diagram (11).

Notice that the distillate composition is controlled by manipulating distillate rate. If the distillate product contains too much heavy key, too high a distillate rate is indicated. As the distillate rate is reduced the liquid level rises in the condenser increasing tower pressure. The pressure controller responds by increasing the reflux rate.

A material balance control scheme can also be set up to control bottoms composition by manipulation of the bottoms rate. In these schemes the heat input at the reboiler is used to control either the liquid level in the tower bottoms or the tower pressure. When material balance control is used it is common to control the composition of the most important product by manipulating the heat balance and control the composition of the other product by manipulating the material balance.

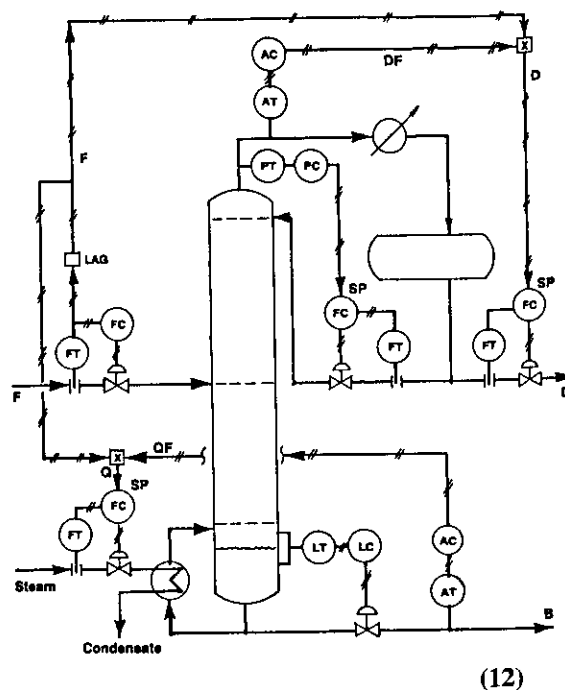
Feed forward control is used to minimize the upsets caused by changes in the feed-rate, composition, temperature. An effective feedforward control system allows the column to operate as closely as possible to product specifications with a minimum of "over fractionation" (exceeding product specifications).

One example of a feed forward system is shown in diagram (12). The feed rate signal is fed forward to both the reboiler (heat input) and distillate rate. Both variables are adjusted based on feed rate changes.

Feedforward control can improve the response of the control system to upsets saving energy and minimizing off-specification fractionation.

Feed composition is generally not monitored in a feedforward control system as composition changes tend to be of lower magnitude and frequency than rate changes. Also, analyzers are more expensive and less reliable than flowmeters.

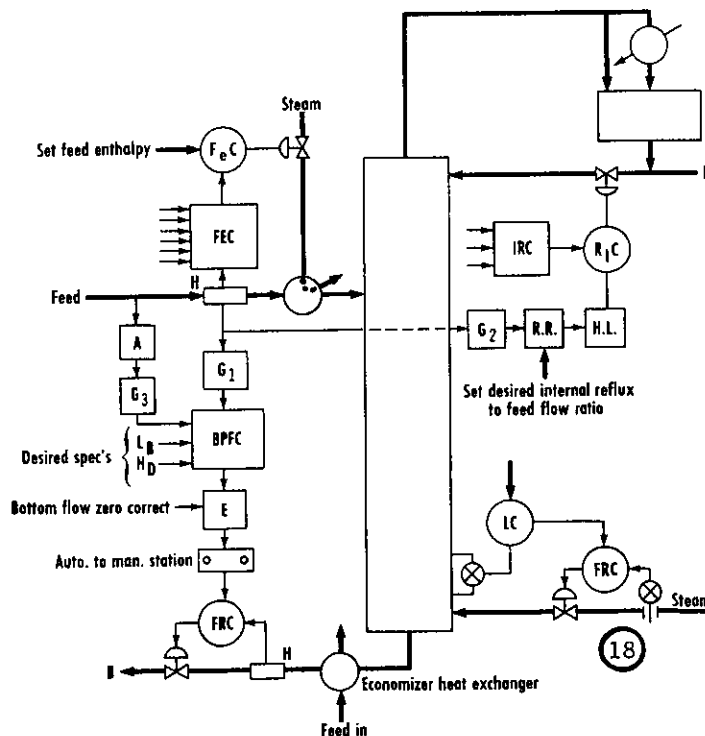
The open loop response of a fractionator to feed changes is not instantaneous. Liquid flow across the trays must change and in a column with 50-60 trays this may induce considerable lag. Temperature changes are subject to lags due to contact efficiency, resistances to heat transfer, etc. When feedforward control is used the feed forward signal is often "dynamically lagged" so that it arrives at the manipulated variable at the correct time and provides the right amount of correction.



Computers are often used to provide this dynamic lagging. This is easily done today with the wide availability of microcomputers. The fractionation process is essentially simulated by the computer and the magnitude and timing of the feedforward signals is computed. The use of computers in fractionation control is "trendy." However, the increased cost and complexity of the control system must be economically justified by lower operating costs or more profitable product yields. This justification is sometimes more imagined than real.

Diagram (13) shows one approach for incorporating a computer into the system. Two-four component concentrations are used for input. The purpose is to maintain constant and/or optimum separation with changes in feed conditions.

BPPC	Bottom product flow computer
H	Linear flow transmitter
A	Analyzer
E	Biasing device
R.R.	Radio device
FEC	Feed enthalpy computer
FeC	Feed enthalpy controller
IRC	Internal reflux computer
RiC	Internal reflux controller
H.L.	High-Low limit

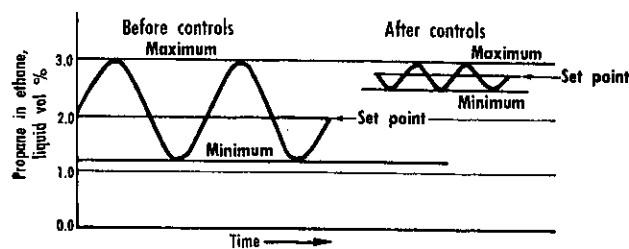


(13) Predictive control system for a distillation column

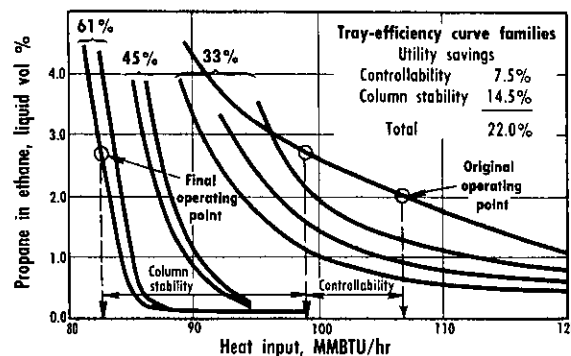
The use of computer controls (like most anything else in life) can be a plus factor if used in moderation and chosen wisely. One of the big savings may be in fuel (utility) savings, in addition to better control.

The figures at right show one example for a deethanizer.^(17,14) The top figure shows the deviation in the volume percentage of ethane in the propane plus bottoms before and after installing computer control. Narrowing the deviation obviously improves controllability and results in greater savings (and ease of operation).

The bottom figure shows a summary of performance for three different overall tray efficiencies 61, 45 and 33%. By moving (and holding) closer to the set point, a 22% savings in utility consumption is achieved. This alone is significant. It also results in greater tray efficiency since the column is more stable.



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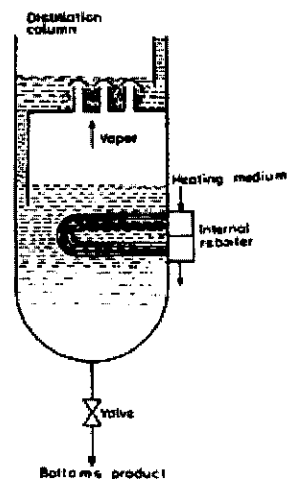
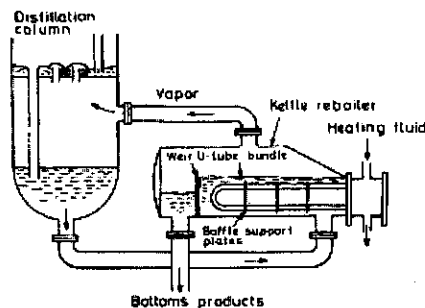
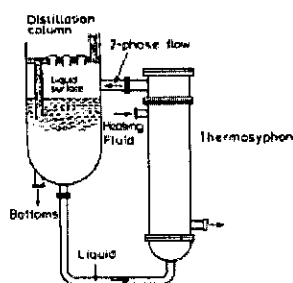
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Condensers and Reboilers

The exact control of condensers and reboilers depends on the type. Condensers may be either the open type which are located in the bottom of a cooling tower and do not contain a shell, or they may be of the closed type which contain a shell and generally are located near the bottom of the fractionator. In condensing a vapor with water, the greatest resistance to heat transfer will be the water film. For this reason, efforts to increase the overall coefficient should center around increasing the heat transfer coefficient on the water side. In the closed type condenser, this can be accomplished by putting water through the tubes. A water velocity of about 1.5 m/s [5 ft/sec] is considered to be about optimum. An average overall coefficient for condensers used on light hydrocarbon fractionators would be $560 \text{ W/m}^2\cdot^\circ\text{C}$ [$100 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$].

The feed preheater may use steam for the heating medium or it may make use of a hot hydrocarbon stream such as the column bottoms. If steam is used, a value of about $330 \text{ W/m}^2\cdot^\circ\text{C}$ [$600 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$] for the overall coefficient would represent a fair average. When using another hydrocarbon liquid stream to heat the feed, the value of U would be about $560 \text{ W/m}^2\cdot^\circ\text{C}$ [$100 \text{ Btu/h}\cdot\text{ft}^2\cdot^\circ\text{F}$]. This value would also represent a fair average for product coolers. When using a liquid hydrocarbon stream, the optimum velocity of the hydrocarbon through the heat exchanger will be between 2-3 m/s.

Reboilers on light hydrocarbon fractionators can be divided into three categories or types: internal reboilers, external "kettle type" reboilers, and external "heat exchanger type" reboilers. Heat exchanger type may be "pump-through" or thermosyphon. In most applications, the "heat exchanger type" reboilers are showing themselves to be superior to the other two types for hydrocarbon fractionation service. All three are shown at the top of the following page. In designing a heat-exchanger type reboiler, the reboiler should be placed below the bottom of the column so that a sufficient head (2-3 m) will be available to provide adequate thermal circulation. Developments in evaporator design and developments in heat exchanger type reboiler design are very much the same since they have approximately the same objective in mind – reduction of required heating surface for a given heat transfer requirement. Trends in design of both are toward long tube, vertical heating surfaces using either natural or forced convection. In a heat exchanger of this type, the boiling film coefficient is increased to a value comparable to the coefficient on the steam side. Δt has a very marked effect on the heat transferred to a boiling liquid, so that $U\Delta t$ is more nearly constant than U itself. In reboiler design, the tendency is to design using a $U\Delta t$ of $32\,000\text{--}39\,000 \text{ W/m}^2$ [$10\,000\text{--}12\,000 \text{ Btu/h}\cdot\text{ft}^2$].



while limiting the Δt to below 40°C [72°F]. The maximum heat transferred per area occurs when the Δt between metal wall and the boiling liquid is from 15-30°C [27-54°F].

It is critical that the reboiler have sufficient vapor disengaging area and sufficient size flanges and return piping. If not, vapor binding may limit capacity instead of area.

For thermosiphon type reboilers, size of the vapor-liquid return line also is critical. The normal liquid velocity in the inlet line is 0.7-2.0 m/s [2.3-6.5 ft/sec]. The velocity in the return line may be estimated by the equation

$$v = \left(\frac{A}{\rho_m} \right)^{0.5} \quad (17.14)$$

Where: A = constant
 ρ_m = mixture density
 v = velocity

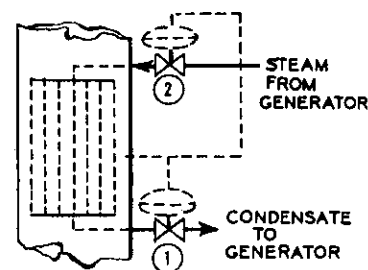
Metric	English
6000	4000
kg/m ³	lbm/ft ³
m/s	ft/sec

Accumulators are designed for a total capacity which will handle the liquid volume of the stream passing through them for from 10 to 30 minutes. Large accumulators offer the disadvantage of requiring more time for change in column overhead to adjust itself. Smaller accumulators are indicated from a process standpoint, particularly when a "flooded" condenser is used.

The heat transfer medium can be a circulating fluid such as hot oil, Dowtherm, Therminol, etc. or steam. Steam provides higher heat transfer coefficient but steam systems are more expensive to build and operate. Steam is popular in facilities where low pressure steam is available from another part of the process such as steam turbine exhaust, or Claus sulfur recovery unit.

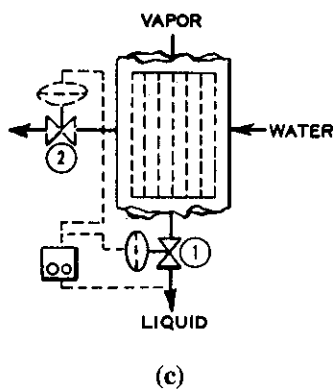
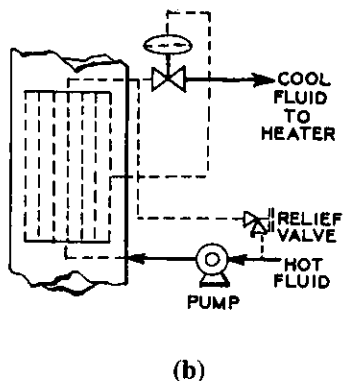
Hot oil systems are the most popular in light hydrocarbon distillation.

Steam controller. – The heating system shown in Part (a) of the figure at the right is simple and transfers a large amount of heat per unit area of coil surface. In most field applications, it is difficult to use a simple, self-actuated, temperature controller that has at least a 10% proportional band. The valve may be located in either position (1) or (2) as shown.



(a)

Position (1) is preferred, for it requires a smaller valve and usually will give slightly more accurate control. In this position, the temperature in the vessel is controlled by the height of condensate in the coil, i.e., changing the area available for transferring heat to the vessel. For example, if the temperature is too low the valve opens wider to drop the level and expose more area. In the event that the heating element is a low coil, position (2) would be necessary.



Hot Liquid. – A typical hot-liquid circulation system is shown in Part (b). The heater may be of any type, but it is often a direct fired heater. In this service it preferably has two thermostats, one for control and one as a high-temperature shutdown. The working pressure is usually about 300 kPa [44 psig].

With a close-clearance centrifugal or a positive-displacement pump, the bypass is essential to keep from overpressuring the system if the control valve closes. When using the latter, the control valve may be placed on the bypass. The proper amount of fluid is then bypassed around the heater to maintain the proper temperature. The circulation rate should be such that this valve normally will be half open, affording temperature control in both directions. If the pressure drop across the heating coil is small, the system shown in Part B is preferred, for a valve on the bypass would have to operate on too small a pressure differential.

Condenser. – Part (c) shows two accepted modifications on a system where a vapor is being condensed, using cold water. The amount of heat transferred may be fixed by controlling the amount of water circulated or by fixing the flow rate and varying the heat transfer surface available. Where the heating coil has sufficient height, the latter method is preferred. The water rate then may be controlled manually.

In most oil field applications, there is no significant difference in the two methods. When using stacked finned-tube exchangers, the valve must be placed in position (2).

OIL STABILIZERS

Stabilization of liquid from crude oil and natural gas liquid separation process is very attractive economically. This is particularly true in the case of high gravity crudes or low temperature separation processes. However, stabilization is not limited to this type of process and is not nearly so widely utilized as it should be. The primary reason for this lack of use seems to be the apparent complexity of this approach compared to stage separation (and, in many cases, mere habit). However, more use is now being made on large oil projects because of higher efficiency than stage separation and the ability to control vapor pressure for tanker shipment.

Figure 17.8 is a view of a typical stabilizer. In some cases of lease operation, the column will operate as a nonrefluxed tower. This type of operation is simpler but less efficient than the refluxed tower operation shown in the inset to Figure 17.8. The nonrefluxed tower should never be used when the feed temperature-pressure relationship exceeds that shown in Figure 17.9. The nonrefluxed tower requires no external cooling source, so it is particularly applicable to remote locations. When a condenser is used in a stabilizer separation, it will nearly always be a partial condenser because of the quantities of methane and ethane that must be removed from the tower feed.

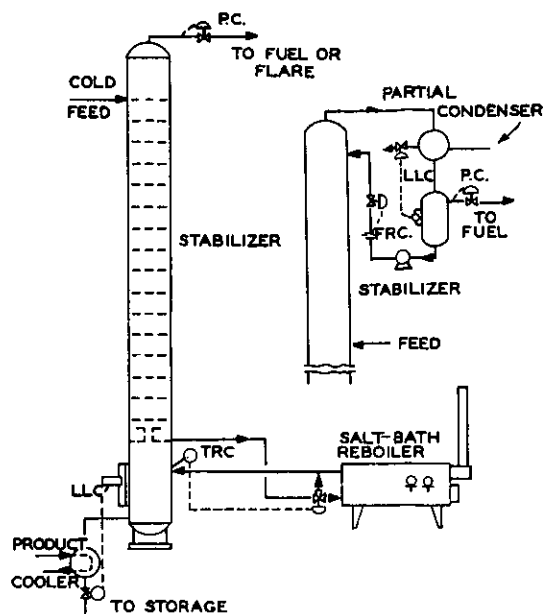


Figure 17.8 Schematic View of Stabilizer

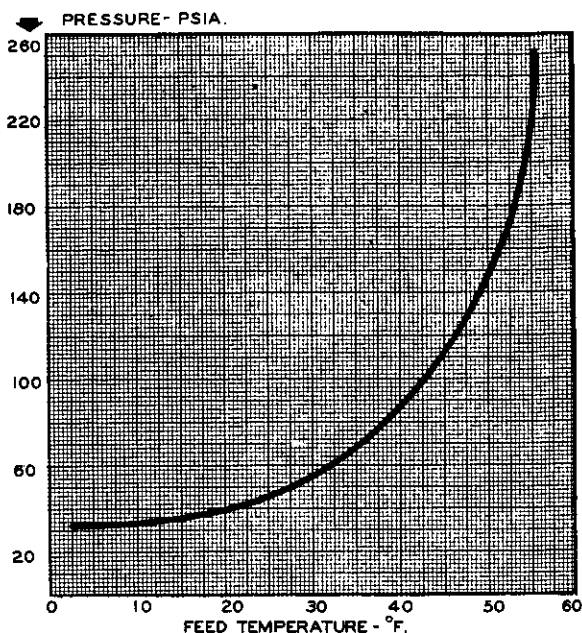


Figure 17.9 Maximum Recommended Feed Temperature to Cold-Feed Stabilizer

A refluxed stabilizer is designed using normal fractionation methods. In general, the specification product will be the bottoms from the tower and in most cases the specification on this will be a vapor pressure limitation. A nonrefluxed tower cannot be designed in the way described for fractionators. There is no external reflux for the tower and so one degree of control over the tower has been lost.

Figures 17.9-17.11 show convenient guides for estimating proper operating range of a nonrefluxed stabilizer. Figure 17.9 shows the maximum recommended feed temperature to a stabilizer as a function of

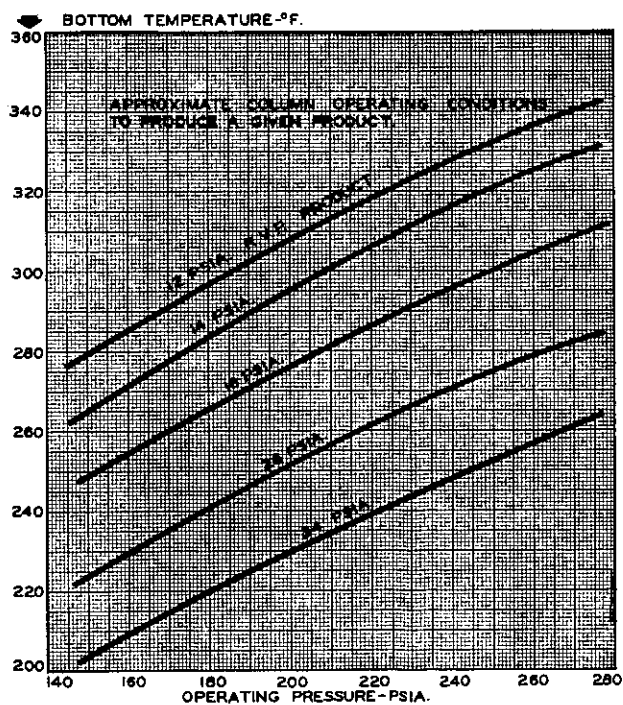


Figure 17.10 Estimation of Proper Bottom Temperature of Stabilizer

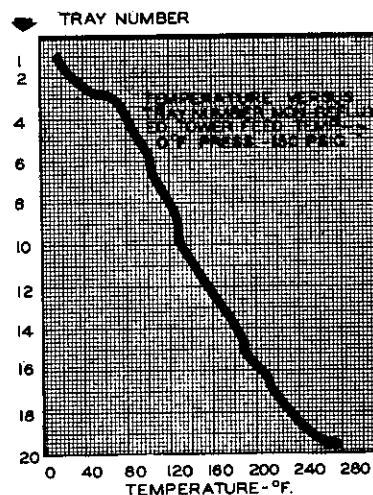


Figure 17.11 Typical Temperature Profile of Cold-Feed Stabilizer

operating pressure. Any lower temperature feed to the column should operate satisfactorily, but the feed to the stabilizer should never be higher than that indicated by the curve in Figure 17.9. An exception to this may be the case where small quantities of liquid are being handled and where first cost is very critical. In these cases, 300-400 kPa [44-58 psia] working pressure nonrefluxed tower with a low pressure steam generator as reboiler can sometimes show economic advantage.

In some cases the reboiler for the stabilizer will be an indirect salt bath heater or a steam-fired heat exchanger. Figure 17.10 shows suggested bottom (reboiler) temperatures for producing a specified Reid vapor pressure product. The temperature for a stable crude oil or a stable stock tank condensate will be in the vicinity of the 12 psi curve.

After the operating temperatures and the pressure have been established through use of Figures 17.9-17.11, the split in the tower must be predicted. Rigorous simulation using a computer is routine today, however, there are several shortcut methods which give reasonable results. One of the most convenient and accurate involves utilization of pseudo-K values for each component between the top and the bottom of the tower. Using this concept, the separation that can be achieved across a nonrefluxed stabilizer can be estimated by use of the pseudo-K values and a simple flash calculation. When using this approach, the following procedure is recommended:

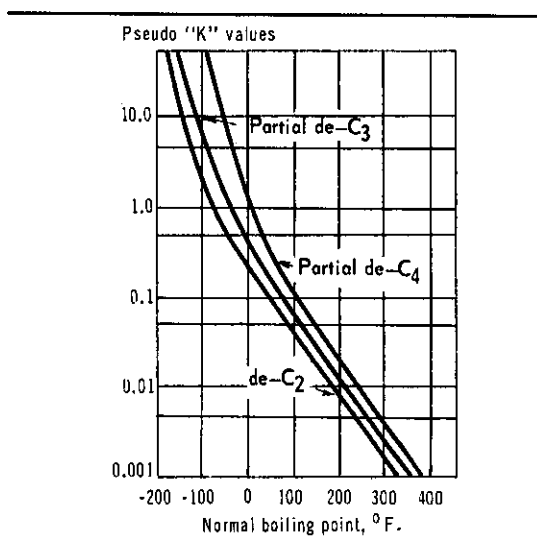


Figure 17.12 Pseudo K Values for Cold-Feed Stabilizers

1. Using Figures 17.9 and 17.10, estimate the operating pressure of the stabilizer.
2. Obtain pseudo-K values from Figure 17.12 for each of the components in the feed mixture.
3. Flash the material to obtain the product split that the tower will make. The vapor from the flash calculation will be the composition of the overhead product and the liquid from the flash will be the composition of the bottom product.
4. Determine the top tower temperature as the dewpoint of the calculated vapor and the bottom tower temperature as the bubblepoint of the calculated bottom product.
5. Make an energy balance around tower to find reboiler heat duty.

The tower will be sized like a regular fractionator.

Use of stabilization will ordinarily enable one to increase recovery and/or crude API gravity, as well as reduce weathering in storage and during shipment and transfer. Stabilization should always be considered an economically attractive alternative to stage separation. In heavy crude areas where "spiking" with light components is common, stabilization of the mix is frequently desirable. Obtaining good mixing when "spiking" is usually much more difficult than one would first anticipate.

Fractionation-type stabilization is being used more extensively offshore in lieu of separator-type stabilization discussed in Chapter 5. Fractionation stabilization is often a more effective method of retaining the C₃-C₅ hydrocarbons in the crude or condensate product while meeting the RVP specification. Other advantages cited include reduced compression pumping costs and better water/oil separation.

Based on computer simulation, Reference 17.15 suggests fractionation-type stabilization be evaluated when

1. If the crude must have an export quality of 50-60 ppm H_2S , maximum.
2. If tanker-quality crude is required (10-12 RVP) and compressor-interstage condensate cannot be exported.
3. If gas cannot be exported via sales line and must be re-injected.
4. If upstream oil-gas separation temperatures are low.
5. If upstream oil-gas separation temperatures are low.

The use of a stabilizer is not recommended if:

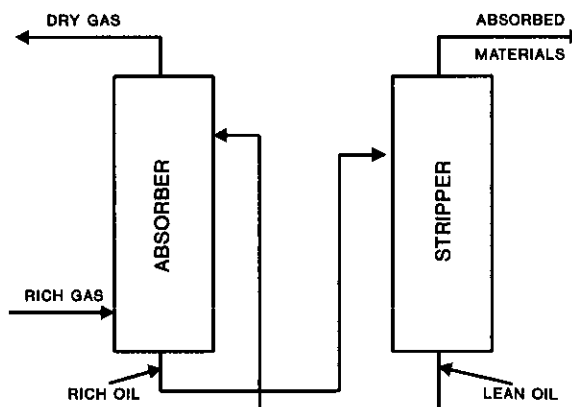
1. A pipeline export crude at an RVP < 12 is not specified.
2. Small amounts of vapor are flowing as may occur in low GOR crudes.

ABSORPTION

Physical absorption is used primarily in gas conditioning for the removal of water, hydrogen sulfide and carbon dioxide. Chapter 18 covers glycol dehydration; Vol. 4 of this series addresses sweetening.^(17.16)

At one point in time, oil absorption plants were the primary source of natural gas liquids. They have been supplanted, generally, by one of the refrigeration processes. Although the discussion that follows is based on lean oil examples, the word "oil" could be replaced with glycol, amine, Selexol, etc.

Rich gas (sometimes called wet gas) enters the bottom of the absorber and flows upward through the absorber counter-current to the lean oil. The absorbable components are transferred from the gas to the oil. The dry gas (sales gas) leaves the top of the absorber. Lean oil enters on the top tray of the absorber and flows downward counter-current to the rich gas, picking up absorbable materials. Rich oil leaves the bottom of the absorber and flows from there to the top of the stripper. In the stripper, heat is added and sometimes a stripping agent such as steam or dry gas. The absorbed material is removed overhead from the stripper and the lean oil from the bottom of the stripper is recycled to the top tray of the absorber. The figure is a block diagram only, and there are heat exchangers, pumps and perhaps even fractionating towers (a demethanizer or deethanizer) that process the rich oil between the absorber and the stripper.



The absorber-stripper combination are "twins." They go together. The absorber recovers components and then the stripper removes them so the oil can return to the absorber in proper condition.

A glycol dehydration plant is an absorber-stripper combination. The principles outlined herein will be re-applied in Chapter 18.

In the countercurrent contact of the gas and oil, a component is absorbed when its "partial pressure" in the gas phase exceeds the "partial pressure" in the liquid phase. (Partial pressure is used here for discussion purposes, the actual driving force is chemical potential.) The oil acts as a means of lowering the vapor pressure of the components by dilution. The leaner the oil is in the components to be absorbed, the more efficient the absorption process. For a given gas rate and composition, the oil circulation rate, oil composi-

tion and number of equilibrium contacts are the prime variables. Oil rate and number of contacts are interdependent because as oil rate increases it remains leaner for a given absorption and less contacts will be needed.

As the components change phase from vapor to liquid, an amount of energy known as the *heat of absorption* is released. In magnitude, it is slightly greater than the latent heat of condensation. The energy released must be absorbed by the oil and gas as they flow through the tower. Consequently, both the dry gas and the rich oil leave at temperatures higher than the rich gas and lean oil. The total heat release is almost proportional to the amount of gases absorbed because the heat of absorption of light hydrocarbons does not vary significantly from component to component. In some cases, the oil must be cooled externally at some point in the absorber to maintain the temperature at the desired level.

The lean oil used in absorbers will have a molecular weight in the range of 100-200, depending on the average tower temperature. In the range of -18°C [0°F], a 120-140 molecular weight oil is commonly used. At temperature near 40°C , the molecular weight will probably be about 180-200. A properly stripped lean oil will contain few components lighter than pentane in measurable amounts.

A number of absorber calculation methods have been proposed and are summarized in various texts, articles, and handbooks. Only one short-cut method and a tray-by-tray method will be considered here. These will suffice for most applications for those charged with specification and operation of absorption facilities.

Nomenclature

In order to simplify the algebra, a slightly different nomenclature is often employed in absorption than that used in standard equilibrium calculations. This is shown in Figure 17.13.

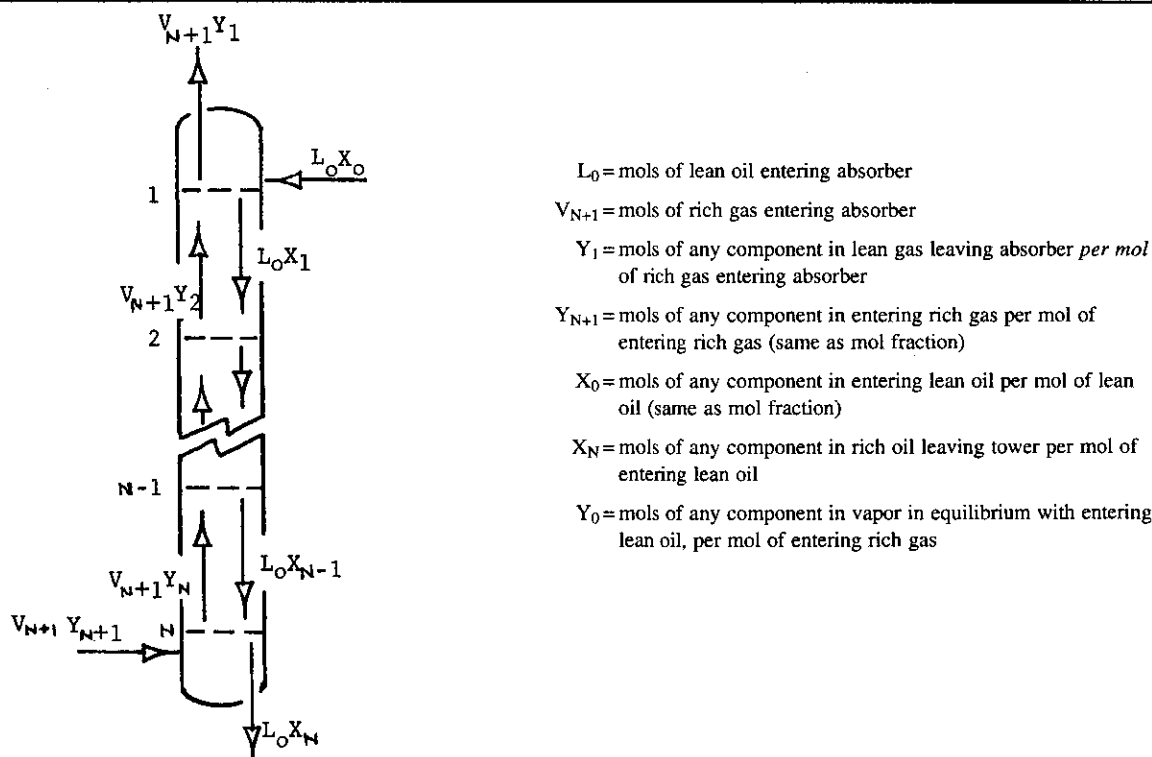


Figure 17.13 Schematic View and Nomenclature for an Oil Absorber

In an absorber the plates are numbered from top to bottom. The bottom plate is plate (N). The subscript on any value is the plate number involved. The absorbent (lean oil) enters from above plate (1) so

it carries the subscript (0). The gas enters from below plate (N) so it carries the subscript (N+1). In some equations the subscript (n) is used. This refers to any general plate (n) from (1) through (N).

Notice that Y is the relative amount of a component in the gas leaving any plate using the inlet gas rate as a basis. The value of X is the relative amount of a component in the liquid leaving any tray using the entering amount of lean oil (L_0) as a basis. Y and X are not mol fractions; they are relative amounts, or ratios, where the denominator is V_{N+1} and L_0 , respectively.

The value of Y_0 is the relative amount of any component in the lean gas leaving if this gas is in equilibrium with the lean oil entering. It is not the case, of course. So, it is merely a fictitious number necessary for the calculation. It would be found by an equilibrium calculation. If the lean oil contains a negligible amount of the component involved, Y_0 is negligible and may be taken as zero for approximate calculations.

Basic Equations

Using the above nomenclature, the equilibrium relationship on any theoretical plate may be written as

$$K_n = \frac{y_n}{x_n} = \left[\frac{V_{N+1} Y_n}{V_n} \right] \left[\frac{L_n}{L_0 X_n} \right] \quad (17.15)$$

A material balance around any plate (n) and the top or bottom of the tower ties together the vapor and liquid streams passing each other between plates. If plates (n-1) and (n+1) are the plates above and below (n) respectively, then for any component

$$\frac{L_0}{V_{n+1}} = \frac{Y_{n+1} - Y_1}{X_n - X_0} = \frac{Y_n - Y_{n+1}}{X_{n-1} - X_n} \quad (17.16)$$

Equations 17.15 and 17.16, together with the corresponding enthalpy balance, can be used as a basis for rigorous tray-to-tray calculations. This becomes a multiple trial-and-error process involving the following steps:

1. Assume a top tray temperature. The entering lean oil temperature plus 3-6°C [5-10°F] is a good guess.
2. Make a dewpoint calculation on the estimated exit gas composition. (The short-cut procedure which follows will provide a good estimate for dry gas composition.) Add enough oil to make the dewpoint calculation. (This also serves to estimate oil losses.)
3. Find composition of liquid leaving top tray using K-values at assumed top tray pressure and temperature.
4. Find composition of vapor leaving Plate 2 from liquid leaving Plate 1 by means of Equation 17.16.
5. Run enthalpy balance around top plate using an assumed vapor temperature from Plate 2. If balance checks, continue a similar calculation for Plate 2. If not, repeat steps 1-5.
6. Proceed down the absorber in this manner until reaching the bottom plate. Then, an overall heat balance must be run. If it checks, solution is complete. If not, the whole procedure must be repeated until the overall heat balance checks.

An aid to estimating liquid and vapor rates for the calculation procedure is to consider that the total absorption is constant on each plate. This leads to the equation

$$\frac{V_n}{V_{n+1}} = \left[\frac{V_1}{V_{N+1}} \right]^{1/N} \quad (17.17)$$

A material balance around Plate (n) and the top of the tower yields

$$L_n = L_0 + V_{N+1} - V_1 \quad (17.18)$$

The temperature on Plate (n) may be estimated by assuming that the temperature change is proportional to the volume of gas absorbed.

$$\frac{T_N - T_n}{T_N - T_0} = \frac{V_{N+1} - V_{n+1}}{V_{N+1} - V_1} \quad (17.19)$$

The above procedure seldom can be justified for routine applications, except on a computer. Certainly, it is too lengthy and time consuming to be used frequently on a hand calculator. Consequently, much shorter and more convenient solutions are desired.

Kremser-Brown Approach

If one assumes that (L/V) does not change very much from plate to plate and that temperature changes are nominal, one can simplify the tower balances. This is a reasonable assumption for most natural gas and glycol absorbers where gas shrinkage is small.

If Equations 17.15 and 17.16 are applied to each plate of an absorber containing a total of (N) plates, Equation 17.20 results.

$$\frac{Y_{N+1} - Y_1}{Y_{N+1} - Y_0} = \frac{A_1 A_2 \dots A_N + A_2 \dots A_N + \dots + A_N}{A_1 A_2 \dots A_N + A_2 \dots A_N + \dots + A_N + 1} \quad (17.20)$$

The absorption factor "A" is defined as $A = L/KV$. For specified Plate (n), $A_n = L_n/KV_n$. Utilizing a mathematical identity, Kremser and Brown simplified Equation 17.20 to:

$$\frac{Y_{N+1} - Y_1}{Y_{N+1} - Y_0} = \frac{A_e^{N+1} - A_e}{A_e^{N+1} - 1} = E_a \quad (17.21)$$

where A is the average of "effective" absorption factor.

Kremser-Brown recommend defining the effective absorption factor (A_e) as:

$$A_e = \frac{L_0}{K V_{N+1}} \quad (17.22)$$

The ratio (L_0/V_{N+1}) is the mols of lean oil per mol of entering gas – a constant for a given calculation. "K" would be the value for the key component at average column conditions – average column temperature and pressure. The key component will be:

The component whose recovery is specified or the component whose "A" is closest to unity where no recovery of a key component is specified.

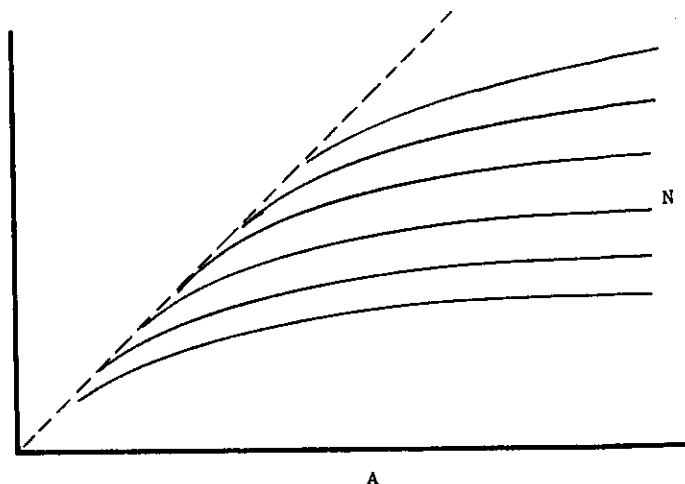
If the lean oil is essentially free of a given component, $Y_0 = 0$ for that component. In this case, the left-hand side of Equation 17.21 represents the fraction of that component absorbed (E_a).

For analyzing an existing tower, the top and bottom temperatures may be averaged to find the average "A." For approximate design, the bottom temperature may be assumed to be about 3-8°C [5-15°F] above that of the entering rich gas. The top tray temperature should be estimated to be 3-6°C [5-10°F] above the entering oil temperature.

One can plot Equation 17.21 in graphical form. The result is a figure possessing the general characteristics shown.

All of the lines representing a constant value of N become coincident with the 45° dashed line representing an infinite number of theoretical plates. For a given value of E_a the combination of A and N used must be to the right of the dashed line for an actual unit.

The lines for N become rather flat as A increases. The most economical unit would be designed to operate a reasonable distance to the right of the dashed line in the curved portion of these lines.



There are three variables that one can control in absorption, assuming the quantity, composition and pressure of the inlet gas are fixed – oil circulation rate, number of theoretical plates and oil temperature. Figure 17.14 is a plot of Equation 17.21. From Figure 17.14, the relationship between absorption factor, number of theoretical plates and fraction absorbed can be obtained for each component. The average "K" should be found by taking the arithmetic average of the lean oil and entering gas temperature and adding 5°C [9°F]. The pressure drop in most absorbers is small and tower pressure may be assumed constant.

If one wishes to find the oil rate for 85% recovery of propane in an absorber having 8 theoretical trays, one would locate 0.85 on the ordinate of Figure 17.14, read horizontally to 8 trays and drop vertically to read the value of A . Knowing K , V_{N+1} , and A , calculate L_0 , the oil rate in mols of lean oil per mol entering rich gas. Now, in similar fashion, if the oil rate is known, the value of A can be determined; at the intersection of the horizontal line from 0.85 and the vertical line from the value of A , the number of theoretical trays necessary for that oil rate can be found.

Note that the lines of constant N in Figure 17.14 are not smooth curves. This is because the scale for A on the abscissa is not uniform. The scale range has been varied to cover wider ranges of A .

Units

The L and V for this correlation are molar rates. Actual units of measurement or specification normally will be in other units and conversion to mols is necessary. The following conversions are rather common. Those for gas are shown for the standard condition specified.

Calculation of V in Mols

Metric: $P_s = 100 \text{ kPa abs. and } T_s = 15^\circ\text{C}$
 $V \text{ in kmol/h} = 1739 (10^6 \text{ std m}^3/\text{d})$

English: $P_s = 14.7 \text{ psia and } T_s = 60^\circ\text{F}$
 $V \text{ in lb mol/hr} = 110 (\text{MMscf/d})$

Calculation of L in Mols

Metric: $L \text{ in kmol/h} = 1000 (\text{m}^3/\text{h})(\gamma)/(\text{Mol Wt})$

English: $L \text{ in lb mol/hr} = 350 (\text{API bbl/hr})(\gamma)/(\text{Mol Wt}) = 62.4 (\text{ft}^3/\text{hr})(\gamma)/(\text{Mol Wt})$

The results of this calculation may be used as a first guess for a trial-and-error solution (such as plate-to-plate) or as an approximate picture of the expected performance.

Example 17.5: An absorber containing 6 theoretical plates uses a lean oil with a relative density of 0.825 flowing at the rate of 1136 m³/d. The molecular weight is 161. The tower pressure is 500 kPa and the average temperature (for calculation) is 30°C. The gas flow rate is 488 500 std m³ per day. Estimate the recovery of each component.

For this case: $L_0 = (1000)(1136/24)(0.825)/161 = 242.5 \text{ kmol/hr}$

$V_{N+1} = (1840)(0.489) = 899.8 \text{ kmol/h}$

So, $(L_0/V_{N+1}) = 242.5/899.8 = 0.27$

A table may now be prepared using these values. The analysis of the gas in question is shown in Column 2.

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
Comp.	Y_{N+1}	K	A	E_a	Mol/h In	Mol/h Absorbed	X_N	Mol/h Overhead	Y_1
Methane	0.83	34.66	0.0078	0.0078	746.83	5.83	0.024	741.0	0.824
Ethane	0.084	6.78	0.040	0.040	75.58	3.02	0.013	72.56	0.081
Propane	0.048	2.04	0.132	0.132	43.19	5.70	0.024	37.49	0.042
i-Butane	0.009	0.81	0.333	0.333	8.10	2.70	0.011	5.40	0.006
n-Butane	0.017	0.59	0.458	0.458	15.30	7.01	0.029	8.29	0.009
i-Pentane	0.004	0.25	1.08	0.885	3.60	3.19	0.013	0.41	—
n-Pentane	0.008	0.19	1.42	0.96	7.20	6.91	0.029	0.29	—
					899.80	34.36	0.1430	865.44	0.962

Column 8 says that 0.1430 mols are absorbed in the tower per mol of entering lean oil. So, the mols of rich oil leaving per mol entering is 1.1430. At the top of the tower $(L_0/V_1) = 242.5/865.44 = 0.28$. At the bottom $(L_0/V_0) = (1.1430)(242.5)/899.8 = 0.31$. This shows that L/V is rather constant. If a second calculation were made, a new L/V of about 0.295 could be used (the average for the top and bottom).

Isopentane is the key component since its value of A is closest to one. It would be used as the key component for any further calculations necessary.

The gas shrinkage was 3.8%, $(1 - 0.962)$. For this low a value, Kremser Brown is a good approximation. In fact, it tends to predict conservative values, i.e., the tower will perform slightly better than predicted.

The recipe for the above example follows:

1. Write analysis of entering rich gas in Column (2).
2. Tabulate K for each component at average column conditions (pressure is substantially constant throughout) in Column (3).
3. Calculate A in Column (4) using Equation 17.22 for known values of L_0 and V_{N+1} .
4. Calculate E_a from Equation 17.21 or Figure 17.14 and tabulate in Column (5). (Assume $Y_0 = 0$).
5. Column (6) is merely Column (2) times the mols per hour of rich gas entering (899.8 kmol/h).
6. Column (7) = Column (5) times Column (6).
7. Each entry in Column (8) = corresponding entry in Column (7) divided by L_0 (242.5 kmol/h).
8. Column (9) = Column (6) minus Column (7).
9. Column (10) = each entry in Column (9) divided by total mols of rich gas entering.

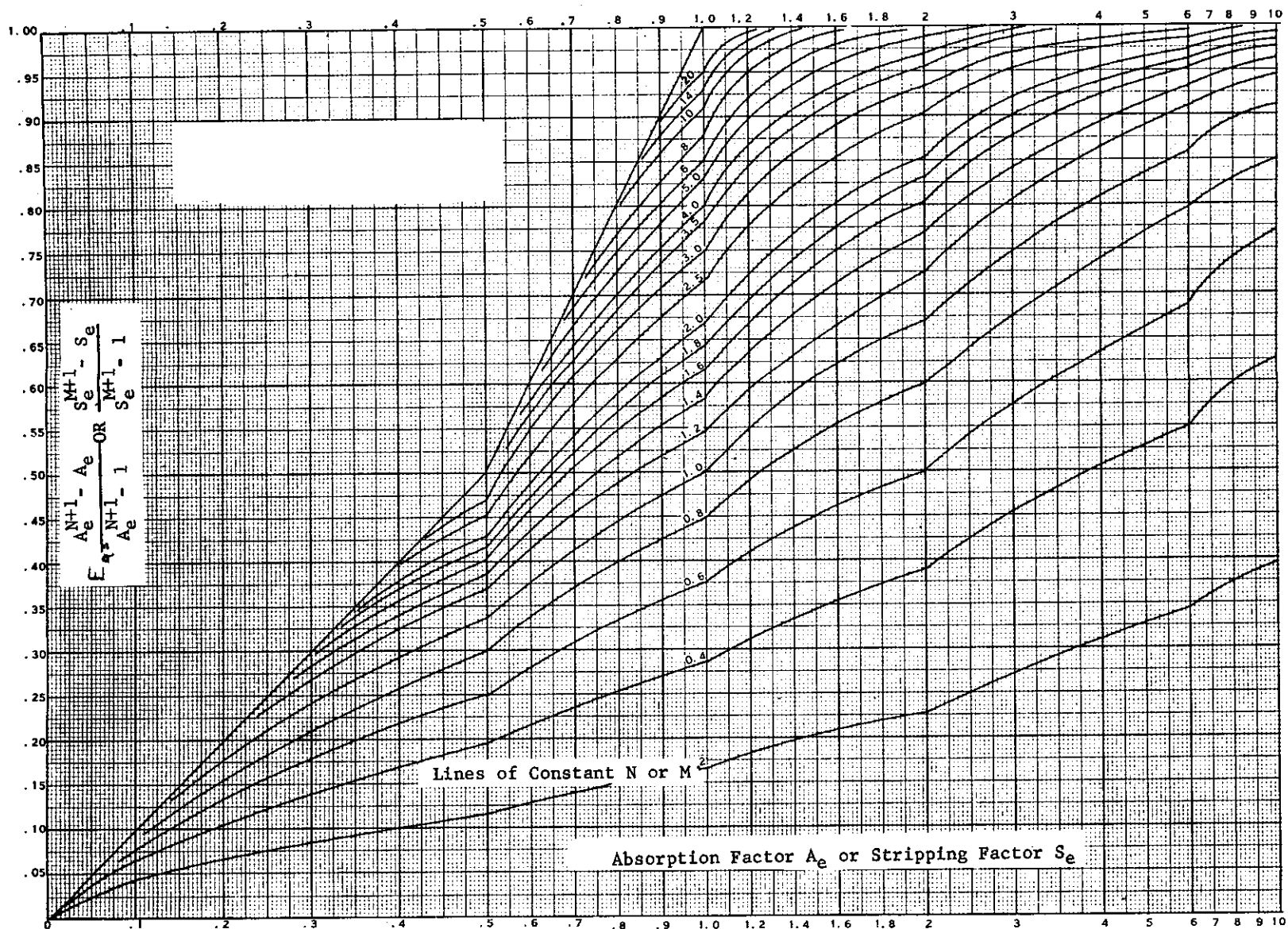


Figure 17.14 Absorption Factor Chart

For the case where recovery of a key component is specified, for a given gas rate and number of plates, the unknown is L_0 . It is found by entering the absorption factor at the known value of E_a , move horizontally to N and then drop vertically to the A axis. Look up K for the key component. Knowing A , K , and V_{N+1} , you can solve for L_0 . This can then be used to complete the table for all other components.

If oil and gas rate is specified, as well as recovery of a key component, one can solve for N , the number of theoretical plates needed. These types of calculations may be used to analyze an existing column to pinpoint any problems or to adapt it for different service conditions.

Most natural gas absorbers contain 7-10 theoretical plates (20-30 actual plates). In planning, the assumption of 8 theoretical plates is a good first guess if a high percentage of propane recovery is desired.

Figure 17.14 shows that as oil rate declines the number of plates tends to infinity. Also, it indicates that more than about 8 theoretical plates gives a minimum decrease in oil rate. The left-hand line in Figure 17.14 represents infinite theoretical plates. Along this 45° line, $E_a = A$. All curves for a finite number of plates coincide with this 45° line at some value of " A ," and follow it at lower values.

As a matter of practical economics one must balance absorber cost with oil rate. The former is a one-time capital investment. The latter involves continuing fuel, pumping and cooling costs. The optimum is usually one which uses the minimum possible oil rate, with an absorber of feasible size, for the desired recovery. Cost is so sensitive to oil rate that rough estimates may be based on it.

The lowest molecular weight oil should be used that conditions permit. This minimizes the mass to be pumped. The limit is fixed by oil vapor pressure and the fact the lean oil should be essentially denuded of the key component.

Since pressure is usually fixed by contract or other practical factors, temperature is the primary process variable. K decreases with decreasing temperature, " A " increases and more recovery is obtained per unit of oil circulated. The use of subambient temperatures therefore decreases the cost of the absorption-stripping portion of the recovery plant, but cooling (refrigeration) costs increase. The optimum temperature may be found by plotting both refrigeration costs and absorption-stripping costs as a function of average absorption temperature. For a given recovery, fractionation costs will be essentially constant. The cost of the absorption section then becomes the cost of the absorber-stripper, oil cooling, pumping and accessories. Most refrigerated absorption plants operate the absorber between -40° and -7°C [-40° and 20°F]. At temperatures in this range, stripping can be obtained by fractionation rather than by the use of steam or a stripping gas. This is an added advantage of subambient absorption temperatures. For recovery of ethane, temperatures down to -50°C [-58°F] are sometimes employed.

Other than mechanical problems, most problems in absorber operation center around oil quality and circulation rate. Proper stripping of the oil is mandatory to minimize losses in the exit gas and insure proper absorption. In most circumstances, the lightest possible molecular weight oil should be used. Vaporization losses limit the oil weight. Also, the oil should contain a minimum quantity of the components for which recovery is desired. In most cases, it should be composed primarily of pentanes and heavier components. Reference 17.17 summarizes desirable absorption oil characteristics.

The procedures outlined determine only the number of theoretical plates. There are many methods available for conversion of theoretical plates to actual plates. As a rule of thumb, an efficiency of 25-40% is close for hydrocarbon absorbers with standard tray designs and in the absence of foaming.

STRIPPER CALCULATIONS

The purpose of the stripper is to remove the absorbed components from the lean oil. Stripping of the absorbed components from the absorption oil is best accomplished by a material immiscible in hydrocarbons because this increases the vaporization tendency of the absorbed components. Steam is commonly used, for it is cheap and readily available. Steam stripping, however, saturates the lean oil with water, requiring that the products or the oil be dehydrated. For this reason, dry gas stripping is sometimes used. This utilizes the leanest gas available (frequently the sales gas). With dry gas stripping, efficiency is somewhat less and condensation of the overhead product is more difficult. When oil molecular weight is below about 140 (low temperature absorption), stripping may be done by fractionation. In this case, the design procedure will follow that outlined in the fractionation section of this chapter.

In a calculational sense, a stripper is simply an upside-down absorber. Trays are numbered from the bottom up with the top plate designated as "M." For convenience, the previous absorption equations will be repeated in proper form for stripping calculations.

$$S = \frac{K V}{L} \quad (17.23)$$

$$\frac{X_{M+1} - X_1}{X_{m+1} - X_0} = \frac{S^{M+1} - S}{S^{M+1} - 1} \quad (17.24)$$

Where: S = stripping factor
 X_1 = mols of component in stripped lean oil leaving bottom of the stripper per mol of rich oil entering the stripper
 X_{M+1} = mols of component in rich oil entering the stripper per mol of rich oil entering the stripper
 L_{M+1} = mols of rich oil entering the stripper
 Y_0 = mols of component in stripping medium per mol of stripping medium entering
 V_0 = mols of stripping medium entering
 X_0 = mols of component in liquid in equilibrium with stripping medium per mol of entering rich oil
 L_0 = mols of lean oil leaving stripper

Equations 17.23 and 17.24 and Figure 17.14 may be used to characterize gas strippers with sufficient accuracy for most operational and planning needs. Glycol and amine type strippers are of the reboiler type and use fractionation and calculations. The current use of purely gas strippers is very limited.

TOWER MECHANICAL DESIGN

A contactor does not know if it is an absorber or a fractionator. It merely reacts to the vapor and liquid loads imposed on it. We will not discuss detailed design herein since that is a specialty area of the vendors and manufacturers. However, the customer must know enough about the design in order to properly prepare specifications and operate the resultant unit.

Mechanical design varies with the choice of the contact device. There are five basic choices: bubble cap, sieve or valve trays, and structured or random packing. There are some general guidelines for the first choice between trayed and packed towers. These are not absolute because in some service they both are suitable, the choice being economic. Generally, though, a packed tower is suitable when: (1) the tower diameter is small, (2) corrosive fluids require special metallurgy, (3) a low pressure drop is needed, (4) the liquid rate is high enough to minimize distribution problems, (5) the depth of packing required does not exceed about 8 meters and/or (6) liquid foaming is not pronounced.

The exact impact of these is different with structured and random packing. For example, it is not a good practice to use random packing with glycol contactors because of the low liquid circulation rate. The problem is liquid distribution. Yet, some vendors will recommend structured packing in this service.

It is difficult to use a trayed tower when the diameter is less than two feet. By the same token, it is difficult to justify packing in large diameter towers.

The pressure drop per theoretical stage is much higher in a trayed tower but this is a factor only in low pressure or vacuum towers. Since it is expensive to manufacture trays from alloy metals, packing offers advantage with corrosive fluids.

The depth of random packing is limited by the crushing or deformation characteristics of the packing. Some plastics are limited to bed depths of 3-4 meters.

When foam is present, it tends to fill the interstices of the packing and enhance flooding. Liquid-solid slurries also are a cause for concern because of the potential for packing plugging. In production operations where salt water may be present, salt plugging is a problem if the temperature is high enough to vaporize the water.

Sieve or valve trays are used in most contactors. They are more efficient than bubble caps when operated at, or near, design capacity. At this capacity they possess about the same efficiency but the sieve tray is less expensive. However, at high turndown the sieve tray efficiency declines faster than either a valve or bubble cap tray because of weeping. So, when turndown is anticipated, the valve tray is the favorite choice, even though it is more expensive.

Bubble caps are now used only in those cases where liquid rates are low and weeping is expected to be a problem. The prime example is the glycol absorber. A weep rate of up to 10% of the liquid circulation rate is satisfactory so long as it is uniform across the tray. When dehydrating "dirty" gas or the glycol contains degradation solids, this rate may be exceeded or weeping may be localized with valve trays. Sieve trays should not be used in glycol service.

Flooding

The terms "flood capacity, flooding point, etc." refer to a condition wherein excess *liquid holdup* occurs in a tower. It is a design limit. When it occurs in operation, excess loss of liquid occurs out of the top of the tower. In addition, process efficiency decreases rapidly.

Flooding can result from several causes. *Jet flooding* results from too large a vapor velocity through the tray or bed. The gas space becomes full of aerated liquid or foam; the liquid cannot flow downward by gravity.

Incorrectly designed trays or downcomers also can cause flooding. Some flooding may result from plugging or obstructions that develop during operations. The tendency to plug may be a practical reason to reject a contact device that otherwise is very suitable. This is true particularly in production operations.

Most will specify a design that does not exceed 75-80% of flood. In some cases, it may be judicious to use not over 50% where flexibility is required. The calculation of flooding capacity is made by the vendor. Most are participants in Fractionation Research, Inc. (FRI), a cooperative of many firms. FRI performs tests and issues standards which often are a part of design specifications. These are considered acceptable standards but some company standards exceed those of FRI.

Tower diameter is affected by both gas and liquid rates, choice of contact device and mechanical design of that device. Tower length is a function of the amount of contact required, the effectiveness of each contact stage and mechanical considerations. The discussion which follows summarizes the major considerations. These are guidelines only and are not offered for use in detailed designs.

TRAY TYPE TOWERS

Diameter

The diameter depends on both liquid and vapor load. The relative effect of each may vary along tower length. This is why some towers have a different diameter at two different points. In most towers, a single diameter is used for economic reasons. Both liquid and vapor requirements are calculated and the larger of the two is used to fix diameter.

Equation 17.25 may be used to estimate vapor rate limitations.

$$w = 3600 K_s [(\rho_L - \rho_g) \rho_g]^{0.5} \quad (17.25)$$

Where: w = vapor mass velocity
 ρ_L = liquid density
 ρ_g = gas density
 K_s = sizing constant

Metric	English
$\text{kg}/(\text{h} \cdot \text{m}^2)$	$\text{lbm}/(\text{h} \cdot \text{ft}^2)$
kg/m^3	lbm/ft^3
kg/m^3	lbm/ft^3
m/s	ft/sec

The value of " K_s " depends on tray spacing and liquid loading but the values below are indicative of those used commonly. Manufacturers will provide specific capacity factors for each proprietary tray.

Service	Tray Spacing					
	Metric			English		
	46 cm	61 cm	76 cm	18 in.	24 in.	30 in.
Absorbers - oil	0.0594	0.0678	0.0719	0.194	0.222	0.236
Absorbers - glycol	—	0.0425	0.0467	—	0.139	0.153
Absorbers - amine	—	0.0297	0.0333	—	0.0972	0.110
Fractionators	0.0372	0.0458	0.0508	0.122	0.150	0.167

Tray spacing of at least two feet is recommended for glycol and amine absorbers. Actually, this spacing also depends on downcomer design.

The tower diameter required for the liquid involves tray design. This is beyond our scope herein, but the following are typical tray characteristics.

Sieve – Hole size varies from 0.64-2.5 cm [0.25-1.0 in.] with about 1.27 cm [0.5 in.] being the most common.

The distance between holes will usually be about 2-3 times the hole diameter.

In most cases, the outlet weir will be about 5 cm [2 in.] high. The inlet weir and downcomer seal must be of a height and area compatible with flow rate.

Valve – The same basic guidelines apply as for sieve trays. The valves vary in size but the most commonly used types are circular (about 5 cm [2 in.] in diameter) or 2.5 cm x 12.5 cm [2 in. x 5 in.] rectangles. About 130 circular valves or about 75 rectangular valves will be used per square meter of active plate area.

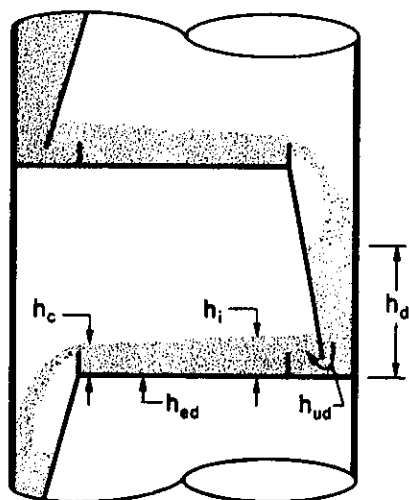
Bubble Cap – The slot (gas) velocity is about 3-6 m/s [10-20 ft/sec], depending on gas density. Since the riser area is slightly larger than slot area, riser velocities will be slightly lower.

Most plates are designed so that the liquid pressure drop across the plate does not exceed 0.35-1.00 kPa [1.4-4.0 in H_2O].

Downcomer Size

Downcomers are sized using velocity, residence time and height of liquid as factors. From these, downcomer area can be found which, in turn, can be used for approximate tower sizing.

Figure 17.15 shows the basic pressure balance around a tray. There must be sufficient head of liquid in the downcomer to flow the liquid across the tray at a proper rate. This height must be less than tray spacing.



Pressure Balance:

$$h_d = (h_t + h_{ud}) \left(\frac{\rho_L}{\rho_L - \rho_V} \right) + h_i + 1.0$$

Where: h_d = downcomer filling, in.
 h_t = $h_c + h_{ed}$
 h_c = clear liquid height, in.
 h_{ed} = dry tray ΔP , in.
 h_{ud} = head loss under downcomer, in.
 ρ_L = liquid density, lb/cu.-ft
 ρ_V = vapor density, lb/cu.-ft
 h_i = tray inlet head, in.

Figure 17.15 Pressure Balance around a Contact Tray

There are many methods for estimating allowable velocity in a downcomer. As a general rule, the velocity should not exceed 0.15 m/s [0.5 ft/sec], regardless of the correlation used. As an approximation, use the lesser of this value and that found from Equation 17.26.

$$v_D = A (\rho_L - \rho_V)^{0.5} (B) \quad (17.26)$$

Where: v_D = downcomer velocity
 ρ_L = liquid density
 ρ_V = vapor density
 A = unit conversion factor
 B = liquid foaming factor

Metric	English
m/s	ft/sec
kg/m ³	lbm/ft ³
kg/m ³	lbm/ft ³
0.0069	0.09
—	—

The values of "B" can be taken as 1.0 for normal light hydrocarbons and 0.70-0.85 for amines and glycols, depending on the amount of foaming.

An alternative approach uses residence time. This will vary from 3-10 seconds, depending on the liquid and its foaming tendencies. The following are representative values.

Amine Type Absorbers	6-8 sec	Glycol Still	5-6 sec
Glycol Absorbers	6-8 sec	Sour Water Stripper	6-8 sec
Oil Absorbers	4-5 sec	Oxygen Stripper	3-4 sec
Amine Still	5-6 sec		

A residence time of around 4 sec is suitable for most hydrocarbon fractionators.

The allowable velocity may be found from residence time by the equation

$$v_d = \frac{h}{(\text{Residence Time})} \quad (17.27)$$

Where: h = height of liquid in downcomer

A given residence time and a given velocity fix downcomer height, which in turn fixes plate spacing. What about the percentage of this height filled with liquid? As a general rule, the following guidelines are realistic.

Pressure, MPa	Max. Liquid Level as % of Tray Spacing
Greater than 2.5	35
0.7-2.5	40
Less than 0.7	50

Once again, foaming is a consideration.

The total downcomer area normally will be 10-12% of total column area unless the liquid rate is unusually low. In no event should the actual area be less than twice the minimum area calculated. Another rule of thumb is that the minimum weir length should be 50% of the tower diameter.

PACKED TOWERS

Packed tower calculations are identical to trayed tower calculations up to the point of tower sizing. The determination of both packed tower diameter and height require special correlations.

Packed Tower Diameter

The diameter of packed towers has historically been calculated by estimating the gas velocity at the *flood point* then sizing the tower so that the actual velocity is 50-60% of the flooding velocity. The flood point can be defined several ways but a widely accepted definition is the point where measured liquid hold-up increase abruptly. It is an unstable condition which will result in poor tower performance and/or liquid carryover. The flood point is a function of liquid rate, packing characteristics, gas and liquid densities and liquid viscosity.

Sherwood^(17.18) was the first to develop a correlation by which flooding velocity could be determined. This correlation was extended to include several curves for various pressure drops through the tower. This model is called the Generalized Pressure Drop Correlation (GPDC) and is widely used today for sizing packed towers. Several versions of the GPDC have been developed. One of the most recent versions is shown in Figure 17.16.^(17.19) Some correlations show an additional curve above the 1.5 in H₂O/ft curve to represent flooding.

$$Y = \frac{G^2 F v^{0.1}}{\rho_g (\rho_L - \rho_g)} \quad X = \left(\frac{L}{G} \right) \left(\frac{\rho_V}{\rho_L} \right)$$

Where: L = liquid mass velocity, lbm/ft²-sec
 G = gas mass velocity, lbm/ft²-sec
 ρ_L = liquid density, lbm/ft³
 ρ_V = gas density, lbm/ft³
 v = liquid viscosity, centistokes
 F = packing factor

The packing factor is determined empirically based on experimental data. Packing factors can be obtained from the packing manufacturer. Table 17.1 shows packing factors published by one packing manufacturer (Norton) for their products. It is important to recognize that packing factors also depend on the GPDC which was used to correlate results. It is always best to check with the a manufacturer to find the GPDC which is consistent with their published factors.

TABLE 17.1
Norton Company - Generalized Pressure Drop Correlation, Jan. 1986 (Packing Factors)

Packing Type	Material	Nominal Packing Size					
		5/8" or #15	1" or #25	1.5" or #40	2" or #50	3" or #70	3.5" or #90
IMTP®	Metal	51	41	24	18	12	
Hy-Pak®	Metal		45	29	26		16
Pall Rings	Metal	70	56	40	27		18
Pall Rings	Plastic	75	55	40	26		17
Super Intalox® Saddles	Plastic		40		28		18
Super Intalox® Saddles	Ceramic		60		30		
Intalox® Saddles	Ceramic		92	52	40	22	
Radchig Rings	1/16" Metal	300	144	93	62	43	

Historically the packing factor was estimated numerically by dividing the specific area of the packing (m^2/m^3 , ft^2/ft^3) by the cube of the fractional void space. These values are reasonable for preliminary sizing and/or loading calculations. Reference 17.20 provides a nomograph for estimating packing factors for various packings.

Figure 17.16 can be used in several ways. The design pressure drop depends on the service. The following values may serve as a guide.

Service	ΔP , in H_2O/ft packing
Absorbers/Regenerators Liquids with foaming tendency Light hydrocarbon distillation	0.25-0.50
Atmospheric and H.P. distillation non-foaming fluids	0.50-1.0
Minimum ΔP	0.05
Maximum ΔP	1.0

For a given service the vendor should be consulted for a recommended ΔP value.

Once the design ΔP has been determined, sizing the tower proceeds as follows:

1. Calculate a value of the abscissa parameter "X" (mass rates for L & G can be used since the area term cancels out)
2. Proceed vertically to the design ΔP curve then horizontally to read a value of ordinate parameter "Y"

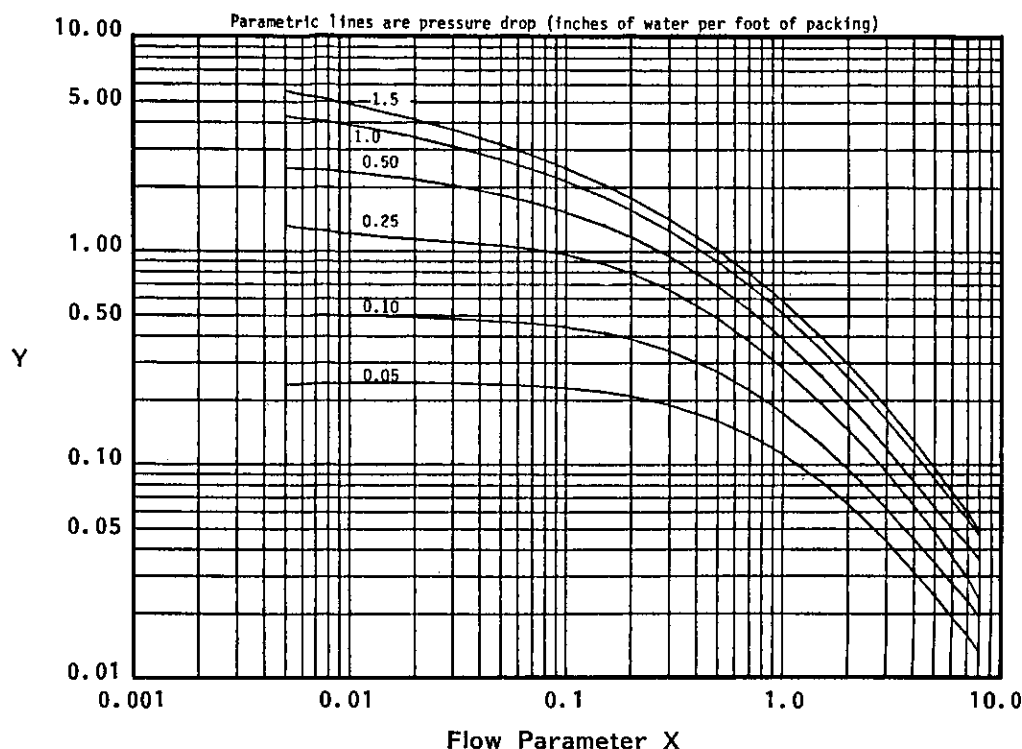


Figure 17.16 Generalized Pressure Drop Correlation

3. From "Y" calculate the gas mass velocity G
4. From G calculate the tower diameter, d , remembering that

$$d = \left(\frac{4m}{\pi G} \right)^{0.5}$$

Where: m = mass flow rate, lbm/sec

Figure 17.16 can also be used to estimate reasonable vapor or liquid loading conditions for an existing tower.

Example 17.6: Calculate the diameter of a glycol contactor, packed with 2 in. Hy-Pak® rings. Gas rate is 0.28×10^6 std m^3/d [10 MMscf/d]. The glycol rate is $0.45 m^3/h$ [2 gpm]. The following data apply:

$$\rho_V = 60 \text{ kg/m}^3 [3.7 \text{ lbm/ft}^3]$$

$$\rho_L = 1120 \text{ kg/m}^3 [69.9 \text{ lbm/ft}^3]$$

$$\nu = 16 \text{ centistokes}$$

$$\gamma_V = 0.62$$

Design the contactor to operate at a pressure drop of 0.25 in H_2O/ft packing.

Calculate "X"

$$m_L = \left| \frac{2 \text{ U.S. gal}}{\text{min}} \right| \left| \frac{9.33 \text{ lb}_m}{\text{U.S. gal}} \right| \left| \frac{1 \text{ min}}{60 \text{ sec}} \right| = 0.31 \text{ lb}_m/\text{sec}$$

Example 17.6 (Cont'd.)

$$m_V = \frac{(10)(3180)(0.62)}{3600} = 5.5 \text{ lb}_m/\text{sec}$$

$$X = \left(\frac{m_L}{m_V} \right) \left(\frac{\rho_V}{\rho_L} \right)^{0.5} = \left(\frac{0.31}{5.5} \right) \left(\frac{3.7}{69.9} \right)^{0.5} = 0.013$$

For $\Delta P = 0.25$ in $\text{H}_2\text{O}/\text{ft}$, "Y" = 1.2

From Table 17.1, $F = 26$

$$Y = 1.2 = \frac{G^2 (26)(16)^{0.1}}{3.7 (69.9 - 3.7)}$$

Solving for G , $G = 2.93 \text{ lbm}/\text{ft}^2\text{-sec}$

$$d = \left(\frac{4m}{\pi G} \right)^{0.5} = \left[\frac{(4)(5.5)}{(3.14)(2.93)} \right]^{0.5} = 1.55 \text{ ft} \quad (\text{use } 18 \text{ in. dia. tower})$$

Tower Height

The height of a packed tower must be sufficient to provide enough contact between the vapor and liquid to give the desired result. In a trayed contact this requires determination of the number of actual trays to be installed. In a packed tower this calculation requires determination of an HETP (Height Equivalent to a Theoretical Plate) or an HTU (Height of a Transfer Unit). These two concepts are related but based on different treatment of the driving force – concentration difference. Some feel the HTU approach is more rigorous, but the HETP approach is frequently used by process engineers because it relates to the equilibrium stage calculations discussed earlier in this chapter.

The actual packing height, h , is then calculated as follows:

$$h = (\text{HTU})(\text{NTU}) \quad h = (\text{HETP})(N)$$

Where: N = number of theoretical stages
 NTU = number of transfer units

The HETP is determined experimentally in laboratory or pilot plant tests. It is a function of packing type, vapor and liquid densities, liquid viscosity and surface tension diffusivity, vapor and liquid loading. References 17.21 and 17.22 provide a review of a few correlations available to calculate HETP.

Few generalized methods for calculating HETP are available in published literature. The packing manufacturer can provide reasonable estimates of HETP for a particular service. For preliminary planning and sizing calculations the following guidelines have proved useful for high efficiency random packing such as slotted rigs.

Packing Size	HETP
2.54 cm [1 in.]	0.46 m [18 in.]
3.81 cm [1.5 in.]	0.66 m [26 in.]
5.08 cm [2 in.]	0.89 m [35 in.]

For glycol dehydration an HETP of 1.5 m [5 ft] can be used to estimate contactor height for both random and structured packing.

Packing performance is often limited by the liquid distribution. At low liquid rates the packing may not be fully irrigated (method) which can have a disastrous effect on performance. When the liquid rate is less than $5 \text{ (m}^3\text{/h)/m}^2$ [(2 U.S. gpm/ft²)] a special distributor design may be required.

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NOTES:

APPENDIX 17A

McCABE THIELE METHOD

This method is a graphical representation of the equilibrium and material balance relationships, and how reflux rate relates to theoretical stages. It was developed for binary mixtures but can be used for multi-component mixtures by the use of key components.

One can write the equilibrium relationship as

$$y = \frac{\alpha x}{1 + (\alpha - 1) x} \quad (17A.1)$$

Where: α = relative volatility, K_L/K_H
 K_L = K for light key component
 K_H = K for heavy key component

This is convenient because relative volatility is relatively constant in most fractionation towers. Figure 17A.1 is a plot of Equation 17A.1.

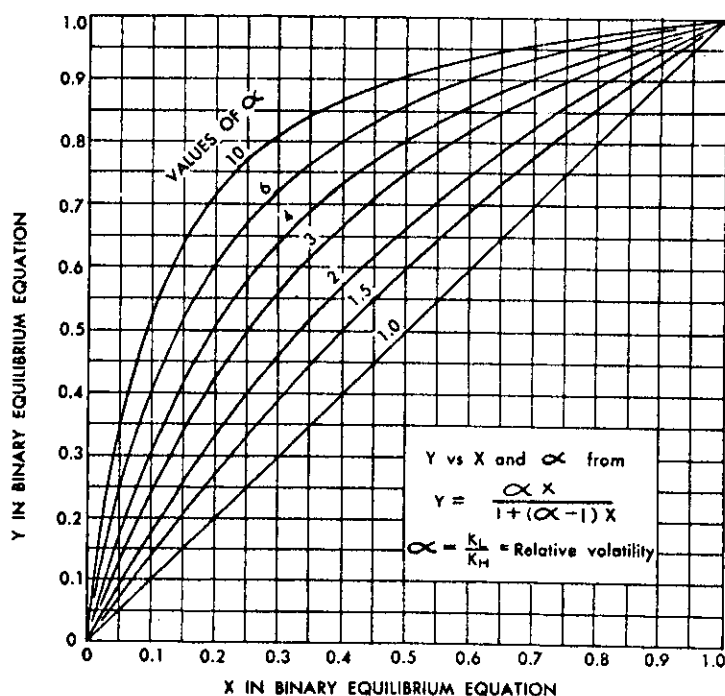


Figure 17A.1 Plot of Equation 17A.1

Material Balance Around Top

A material balance below plate "n" and around the top of the column (as shown by Envelope A in Figure 17.5) relates flow rates of vapor and liquid streams below plate "n."

$$V_{n+1} = L_n + D \quad (17A.2)$$

A material balance (using the same envelope) for one component is

$$(V_{n+1})(y_{n+1}) = L_n x_n + D x_D \quad (17A.3)$$

Rearranging,

$$y_{n+1} = \left(\frac{L_n}{V_{n+1}} \right) (x_n) + \left(\frac{D}{V_{n+1}} \right) (x_D) \quad (17A.4)$$

Where: V_{n+1} = mols vapor entering plate "n"
 L_n = mols liquid leaving plate "n"
 D = mols distillate overhead
 y_{n+1} = mol fr component in vapor entering plate "n"
 x_n = mol fr component in liquid leaving plate "n"
 x_D = mol fr component in overhead product
 L = mols reflux

The basic assumption of the McCabe-Thiele procedure is that the liquid overflow from plate to plate is constant. If we look at Equation 17A.2 and assume that L has the same magnitude for all values of n , we see that V_{n+1} will also be the same for all values of n . Equation 17A.3 then takes the form $y = mx + b$. If plotted on coordinates of y and x , this will be the equation of a straight line with a slope of "m" (L/V) and a y -value at $x = 0$ of Dx_D/V . When plotted on Figure 17A.2, Equation 17A.4 is a straight line located as shown.

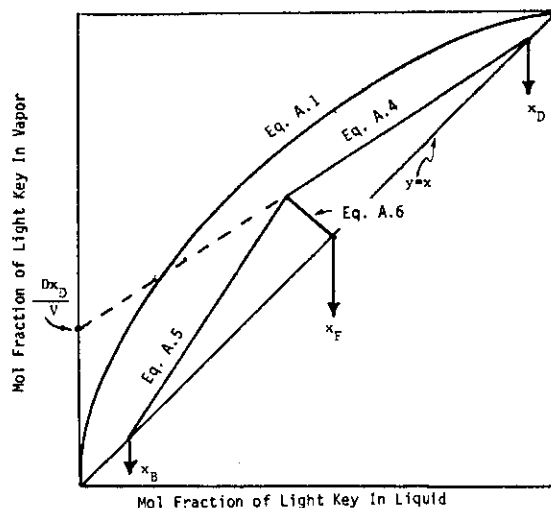


Figure 17A.2 Construction of McCabe-Thiele Diagram

Material Balance Around Bottom

In similar fashion, a material balance around the stripping section of the column (Envelope A in Figure 17.5) will give the material balance line for the stripping section.

$$x_m = \left(\frac{L_m}{V_{m-1}} \right) (x_m) - \left(\frac{B}{V_{m-1}} \right) (x_B) \quad (17A.5)$$

Where: L_m = mols liquid leaving plate "m"
 V_{m-1} = mols vapor leaving plate "m-1"
 B = mols bottom product
 x_m = mol fr of component in liquid from plate "m"
 x_B = mol fr of component in bottoms
 y_{m-1} = mol fr of component in vapor from plate "m-1"

Assuming constant molal overflow in the stripping section leads to the same line of reasoning for the rectifying section. The resulting straight line on y-x coordinates is shown in Figure 17A.2.

The two straight lines represented by Equations 17A.4 and 17A.5 are termed the "*operating*" lines for the rectifying and stripping sections of the column, respectively. They are given this title because their location (slope) is governed by the amount of reflux (L) pumped back to the column – a function of the manner in which the column is operated.

Effect of Feed Condition

A combined heat and material balance around the feed plate may be written in the form

$$x_f = \left(\frac{q}{q-1} \right) (x_f) - \left(\frac{x_F}{q-1} \right) \quad (17A.6)$$

Where: q = total heat needed to convert one mole of feed into saturated vapor divided by the molal latent heat
 x_F = mol fr of component in feed
 x_f = mol fr of component in liquid leaving feed plate
 y_f = mol fr of component in vapor leaving feed plate

Consideration of Equation 17A.6 will make clear that it also is a straight line when plotted on y-x coordinates. It has a slope of $(q/q-1)$. Also, when x is equal to the feed composition (x_F), y also has the value of x_F ($y = x_F$). The behavior of the so-called " q " line around the point $x = x_F$, $y = x_F$, is shown in Figure 17A.3.

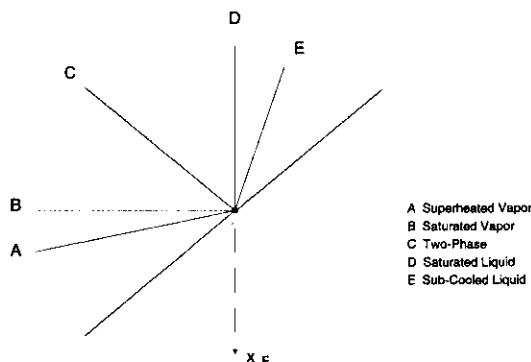


Figure 17A.3 Effect of Feed Condition On "Q" Line

The sequence of steps the engineer should take in constructing a McCabe-Thiele diagram is outlined below.

1. With knowledge of feed composition and condition and the products desired, make product splits for the tower and complete the material balance.
2. Using Equation 17A.1, calculate several points on the equilibrium curve. For an assumed value of x , the corresponding value of y is calculated.
3. Draw the equilibrium curve (for the more volatile component) and the 45° line as shown in Figure 17A.4.

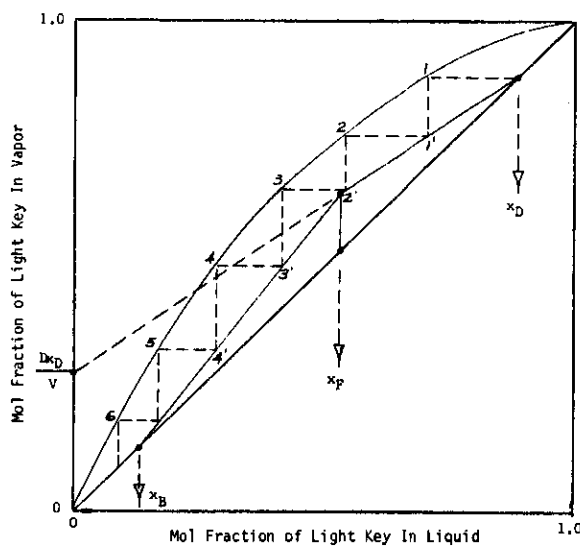


Figure 17A.4 McCabe-Thiele Diagram at a Finite Reflux Ratio

4. Mark x_B , x_D , and x_F on the x -axis and the 45° line.
5. Draw the rectifying section operating line (Equation 17A.4) as a straight line between the points ($x = x_D$, $y = y_D$) and ($x = 0$, and $y = Dx_D/V$).
6. Draw the "q" line as a straight line from the point ($x = x_F$, $y = x_F$) with a slope equal to q divided by $q - 1$.
7. Draw the stripping section operating line as a straight line from the intersection of the rectifying section of operating line and the "q" line to the point $x = x_B$ and $y = x_B$.
8. Step off the number of theoretical plates required to make the specified separation. This is done as shown in Figure 17A.4 by starting at the point representing the distillate composition (x_D) and alternating horizontal and vertical straight lines until the value of the bottoms composition (x_B) is reached.

This stepwise process is equivalent to calculations carried out tray-by-tray. The numbered points (1, 2, etc.) on the equilibrium curve represent the step of determining the liquid in equilibrium with a given vapor. Point 1, for example, represents (by horizontal line to the y -axis) the vapor leaving tray 1 and (by a vertical line drawn to the x -axis) the liquid in equilibrium with that vapor. The numbered points on the operating lines (1', 2' etc.) represent (by a horizontal line to the y -axis) the composition of the vapor leaving Plate 2 and (by a vertical line to the x -axis) the liquid leaving Plate 1. The operating line represents a material balance around the rectifying (stripping) section and must be used to determine the composition of passing streams.

The limits of column operation, minimum plates and minimum reflux, can be illustrated very clearly through use of the McCabe-Thiele diagram.

Consider the overall material balance for the rectifying section (Equation 17A.2). As L_n increases with respect to D , V_{n+1} approaches the value of L_n . When $D = 0$, $V_{n+1} = L_n$, but L_n can never be greater than V_{n+1} . This means that the rectifying section operating line (Equation 17A.4) can never be greater than unity. The same line of reasoning applies to the stripping section operation line (Equation 17A.5). This means that when $D = 0$ (and $B = 0$) the operating line for both sections of the column coincides with the 45° line. This condition is known as "total reflux" and under this condition each plate in the column makes the maximum possible separation. Consequently, the minimum number of plates for the specified separation will occur at total reflux. This situation and the method of stepping off trays at total reflux is illustrated in Figure 17A.5.

Consider what happens when one of the operating lines intersects the equilibrium curve. As we are stepping off theoretical trays and draw closer to the point of intersection, the separation per tray becomes smaller and an infinite number of theoretical trays is required to actually reach the point of intersection. This will be true any time an operating line intersects the equilibrium curve. The one of particular interest to us is the time at which the "q line," the rectifying section operating line, the stripping section operating line and the equilibrium curve intersect at a common point. Under this condition, only with an infinite number of theoretical plates in the column would we be able to make the specified separation of the feed into the distillate and bottom products. The construction of the McCabe-Thiele diagram under conditions of minimum reflux is shown in Figure 17A.6. The minimum reflux rate may be determined from the slope (L/V) of the rectifying section operating line or from its intercept (Dx_D/V) with the y-axis.

Although this method is not used for regular calculations, it is an excellent instructional tool to show the effect of changing operating variables on tower performance.

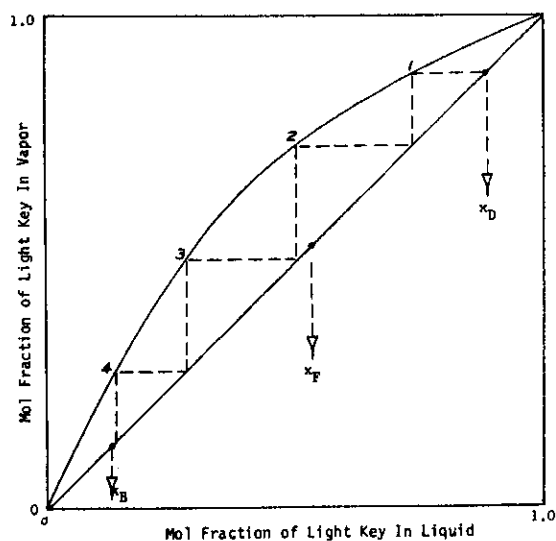


Figure 17A.5 McCabe-Thiele Diagram at Total Reflux

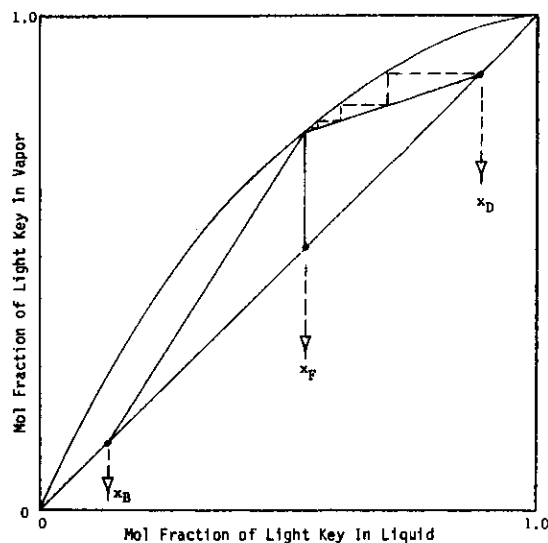


Figure 17A.6 McCabe-Thiele Diagram at Minimum Reflux

APPENDIX 17B

SHORT-CUT UNDERWOOD METHOD^(17.5)

This method uses the equation

$$\alpha_f \left[\frac{(L + D) x_{cf} + (q - 1) D x_{cD}}{(L + D) x_{df} + (q - 1) D x_{dD}} \right] = \frac{L x_{cf} + q D x_{cD}}{L x_{df} + q D x_{dD}} \quad (13B.1)$$

- Where:
- α_f = ratio of K 's for the key components at the feed plate temperature
(temperature of feed at its bubblepoint would be a fair approximation of the feed plate temperature)
 - L = mols of liquid leaving any plate above the feed at minimum reflux
 - D = mols of overhead product
 - x_{cf} = composition, in mol fraction or mol percent, of the *light key* component in *feed*
 - x_{df} = composition, in mol fraction or mol percent, of the *heavy key* component in *feed*
 - x_{cD} = composition, in mol fraction or mol percent, of the light key component in overhead product
 - x_{dD} = composition, in mol fraction or mol percent, of the heavy key component in overhead product

The above equation tends to give high values of minimum reflux. In some cases, the value found may be considered as a minimum in the actual range of reflux rates considered.

If the feed is at its bubblepoint, $q = 1$, this equation is simplified since some terms fall out. This is the case in our continuing example.

For Case I, Equation 17B.1 may be solved for L , where $\alpha_f = 1.81$

$$(1.81) \left(\frac{8.9}{63.3} \right) = \left(\frac{8.9 L + (7.8)(92.3)}{63.3 L + (7.8)(5.1)} \right)$$

$$16.11 L + 10.12 = 8.9 L + 720, \quad L = \frac{709.9}{7.21} = 98.5 \text{ mols/100 mols feed}$$

If we write a material balance around the top of the tower, $V = L + D$ or $V = 98.5 + 7.8 = 106.3$ mols/100 mols feed. Thus,

$$(L/V)_m = 98.5/106.3 = 0.927, \quad (L/D)_m = 98.5/7.8 = 12.63$$

By the other Underwood method, $L = 79.8$ mols/100 mols feed, which is usually rather close to what one would obtain from a more complex model. So, Equation 17B.1 in this example gives results which are high by 23%.

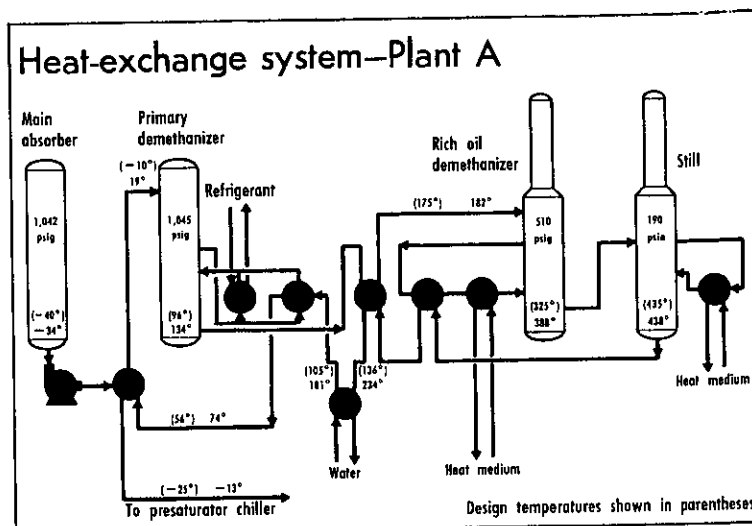
APPENDIX 17C

Suggested Reasons for Deviation of Plant Performance

1. Methane K values used in absorber calculation were too low.
2. Total overhead vapors from primary and rich oil demethanizers were 14% less than design, although 27% more lean oil than calculated was used.
3. Higher lean oil circulation caused the following:
 - a. Higher fuel gas consumption (17%)
 - b. Higher steam consumption (29%)
 - c. Higher cooling tower duty (30%)
 - d. Higher electrical power consumption (17%)
4. The inlet gas—gas heat exchangers did not perform as predicted, as shown below:

	Design	Test
Inlet gas in, °F	84	89
Inlet gas out, °F	-17	-6
Residue gas in, °F	-33	-31
Residue gas out, °F	74	72
MTD, °F	12.8	20.6
Transfer rate, Btu/hr-sq ft.°F	74.5	54.4

5. Many temperatures had to be changed from design conditions to alleviate the above. This change in turn caused further change from design conditions

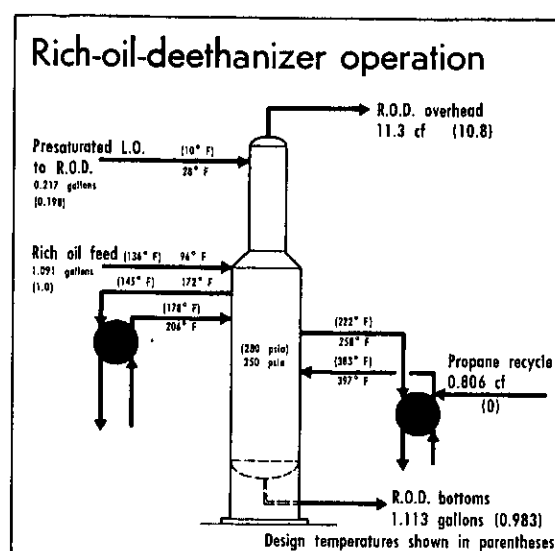
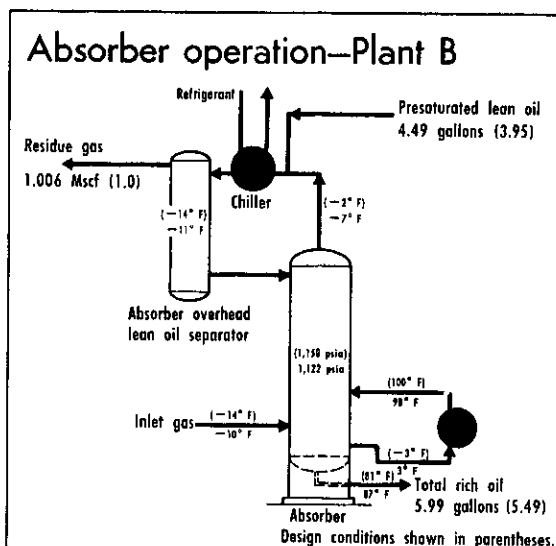


Summary of results, Plant A

Operating conditions	Design	Test
INLET GAS		
Volume (design = unity)	1.0	0.997
Temperature, °F	84	89
Pressure, psig	1,050	1,037
Propane plus content, gal/Mcf	0.614	0.705
ABSORPTION OIL		
Molecular weight	100	103
Gallons/mole	16.29	16.70
Quantity, gal/Mcf of inlet gas		
Main absorbers	1.70	2.16
Rich-oil demethanizer	0.35	0.64
Total	2.05	2.80
Ambient air temperature, °F	100	80
PERFORMANCE		
Ethane recovery, %	25.0	17.4
Propane recovery, %	94.0	92.9
iC ₄ loss to residue gas, gal/MMcf	1.4	13.8
Fuel gas consumption, % of inlet gas	1.1	1.3
Steam consumption, lb/Mcf of inlet gas	8.0	10.4
Electric power usage, kw/MMcf of inlet gas	1.52	1.78
Cooling water, duty, MBTU/Mcf of inlet gas	7.2	9.5
Primary demethanizer overhead, % of inlet gas	6.75	5.18
Rich-oil demethanizer overhead, % of inlet gas	2.47	2.82
	9.22	8.00

Design and test compositions, Plant A—

Component	Inlet gas mole %		Total absorber rich oil mole %		Primary demethanizer overhead—mole %		Rich-oil demethanizer bottoms—mole %	
	Design	Test	Design	Test	Design	Test	Design	Test
N ₂	0.47	0.39	0.09	0.04	0.21	0.14	0.28	0.31
CO ₂	0.98	1.00	1.12	1.55	1.48	1.65	0.32	0.21
C ₁	92.83	92.18	51.55	43.64	91.63	90.06	12.59	8.14
C ₂	3.78	4.24	10.80	10.00	5.37	6.45	14.69	12.12
C ₃	1.16	1.18	7.36	7.37	1.03	0.99	3.33	3.25
iC ₄	0.25	0.30	1.62	1.99	0.12	0.27	3.20	2.68
C ₄	0.24	0.26	1.55	1.64	0.10	0.16	1.26	1.24
iC ₅	0.10	0.11	0.61	0.75	0.02	0.08	0.88	0.70
C ₅	0.07	0.06	0.42	0.42	0.01	0.04	1.72	2.72
C ₆ +	0.12	0.28	0.81	1.65	0.03	0.16	61.73	68.63
Lean oil			24.07	30.95				
	100.00	100.00	100.00	100.00	100.00	100.00	100.00	100.00



Summary of results, Plant B

Operating conditions	Design	Test
INLET GAS		
Volume (design = unity)	1.0	1.006
Temperature, °F	80	85
Pressure, psig	1,135	1,128
Propane-plus content, gal/Mcf	0.806	0.780
ABSORPTION OIL		
Molecular weight	130-135	139
Gallons/mole	20.5-21.3	21.64
Quantity, gal/Mcf of inlet gas		
Main absorbers	3.37	3.87
Rich-oil demethanizer	0.93	1.03
Total	4.30	4.90
Ambient air temperature, °F	100	76
Performance		
Propane recovery, %	90.0	78.6
Fuel gas BTU usage, % of inlet gas	1.8	1.9
Temperature of gas to absorber, °F	-14	-10
Temperature of presaturated oil to absorbers, °F	-14	-11

Suggested Reasons for Deviation of Plant Performance

1. Higher gas and lean oil temperatures than calculated.
2. Higher molecular weight lean oil.
3. Higher propane losses in rich oil deethanizer.
4. Miscalculation of lean oil circulation rate.
5. The rich oil—lean oil heat exchanger system did not have the flexibility needed to compensate for the increased heat input to the reboiler of the rich oil deethanizer necessitated by having to raise its temperature above design.
6. Propane retention errors in the rich oil deethanizer probably resulted from uncertainty in selection of "K" and enthalpy values.
7. Liquid could bypass the absorber reboiler by dumping through the sieve trays used. (Was corrected by converting bottom tray to chimney tray.)
8. The inlet gas could not be chilled to -14°F because thermosiphon system in freon chillers did not work properly because of excess pressure drop.

Inlet gas and rich-oil streams, Plant B

	Design		Test	
	Inlet gas	Absorber rich oil	Inlet gas	Absorber rich oil
C ₁ + inerts	7,833.01	312.30	7,914.30	358.35
C ₂	312.35	114.91	283.75	89.24
C ₃	103.56	99.09	95.84	83.82
iC ₄	14.20	14.19	19.75	18.87
nC ₄	36.74	36.74	32.75	31.45
iC ₅ +	51.78	51.78	51.32	48.46
Lean oil		508.97		557.82
	8,351.64	1,137.98	8,397.71	1,188.01

Percent C ₃ in inlet recovered in absorber rich oil	95.7	87.4
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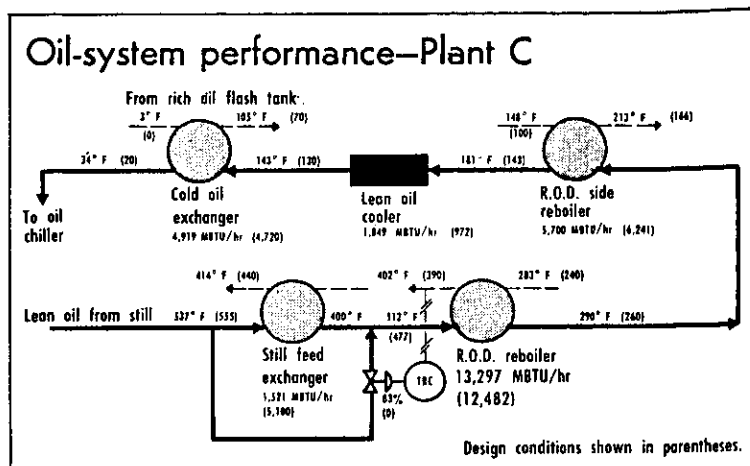
Rich-oil deethanizer streams, Plant B

	Design		Test	
	Rich oil feed	Bottoms	Propane recycle	Bottoms
C ₁ + inerts	312.30		358.35	
C ₂	114.91	2.30	89.24	5.32
C ₃	99.09	93.29	83.82	108.30
iC ₄	14.19	14.19	18.87	18.67
nC ₄	36.74	36.74	31.45	31.15
iC ₅ +	51.78	51.78	48.46	48.16
Lean oil	508.97	649.20	557.82	705.64
	1,137.98	847.50	1,188.01	917.24

Percent C ₃ in rich-oil feed recovered in bottoms	94.2	89.8
Ratio of C ₂ to C ₃ in bottoms	0.0247	0.0491

Suggested Reasons for Deviation of Plant Performance

1. Inlet gas temperature of 90°F instead of 80°F.
2. Inlet gas richer than predicted.
3. Higher recompressor discharge temperature than predicted for gas recycled back into inlet gas (eliminated by air cooler).
4. Higher lean oil temperatures to lean oil chiller.
5. General failure of gas-gas heat exchanger to perform as predicted, as shown below:



Summary of results, Plant C-

	Design	Test
Inlet gas in, °F	80	96
Inlet gas out, °F	20	38
Residue gas in, °F	0	15
Residue gas out, °F	70	83
MTD, °F	14.5	17.5
Transfer rate, Btu/hr·sq ft·°F	71.6	53.8

6. 83% of the lean oil had to be bypassed around the still-feed heat exchanger to maintain proper rich oil deethanizer bottom temperatures. This introduced control problems on maintaining stable bottom temperatures.

Operating conditions	Design	Test
INLET GAS		
Volume (design = unity)	1.0	0.996
Temperature, °F	80	90
Pressure, psig	1,085	1,066
Propane plus content, gpm	0.783	1.136
ABSORPTION OIL		
Molecular weight	135	139
Gallons/mole	20.68	21.86
Quantity, gal/Mcf of inlet gas:		
Main absorber	3.35	3.40
Rich-oil deethanizer	0.72	0.77
Total	4.07	4.17
Ambient air temperature, °F	100	91
Performance		
Propane recovery, %	75.0	64.6
Lean-oil loss to residue gas, gal/MMcf	5	11
Fuel-gas usage, % of inlet gas	1.25	1.68
Temperature of gas to absorber, °F	0	+13
Temperature of oil to absorber, °F	0	+15
Refrigerant rate to chillers, lb/Mcf of inlet gas	21.0	26.4
Refrigerant temperature to chillers, °F	-10	-2

NOTES:

18

GLYCOL DEHYDRATION

A number of liquids possess the ability to absorb water from gas. Yet, there are very few which meet the criteria for a suitable commercial process: are highly hygroscopic, do not solidify in a concentrated solution, are noncorrosive, do not form precipitates with gas constituents, are easily regenerated to a high concentration, can be separated easily, are essentially nonsoluble in liquid hydrocarbons, and are relatively stable in the presence of sulfur compounds and carbon dioxide under normal operating conditions.

Several of the glycols come the closest to meeting all of these criteria. Diethylene (DEG), triethylene (TEG) and tetraethylene (TREG) glycols all possess suitable traits. However, almost 100% of the glycol dehydrators use TEG.

DEG is somewhat cheaper to buy and sometimes is used for this reason. But, by the time it is handled and added to the units there is no real saving. Compared to TEG, DEG has a larger carry-over loss, offers less dewpoint depression and regeneration to high concentrations is more difficult. For these reasons, it is difficult to justify a DEG unit, although a few are built each year.

TREG is more viscous and more expensive than the other processes. The only real advantage is its lower vapor pressure which reduces absorber carry-over loss. It may be used in those relatively rare cases where glycol dehydration will be employed on a gas whose temperature exceeds about 50°C [122°F].

This chapter will concentrate on TEG, even though property data are shown in Appendix 18A for the several glycols. Some of the system characteristics apply also for all glycols.

THE BASIC GLYCOL DEHYDRATION UNIT

Figure 18.1 shows the basic glycol unit, regardless of the glycol used. Not shown is a full size separator ahead of the absorber, an essential piece of equipment. Also not shown is any cooling equipment that may be a part of the dehydration processes. When it is possible to cool the entering wet gas with air or suitable water ahead of the absorber, do so. Such cooling is the least expensive form of dehydration.

The entering wet (rich) gas, free of liquid water, enters the bottom of the absorber (contactor) and flows countercurrent to the glycol. Glycol-gas contact occurs on trays or packing. Bubble cap trays have been used historically but structured packing is more common today. The dried (lean) gas leaves the top of the absorber.

The lean glycol enters on the top tray or at the top of the packing and flows downward, absorbing water as it goes. It leaves rich in water.

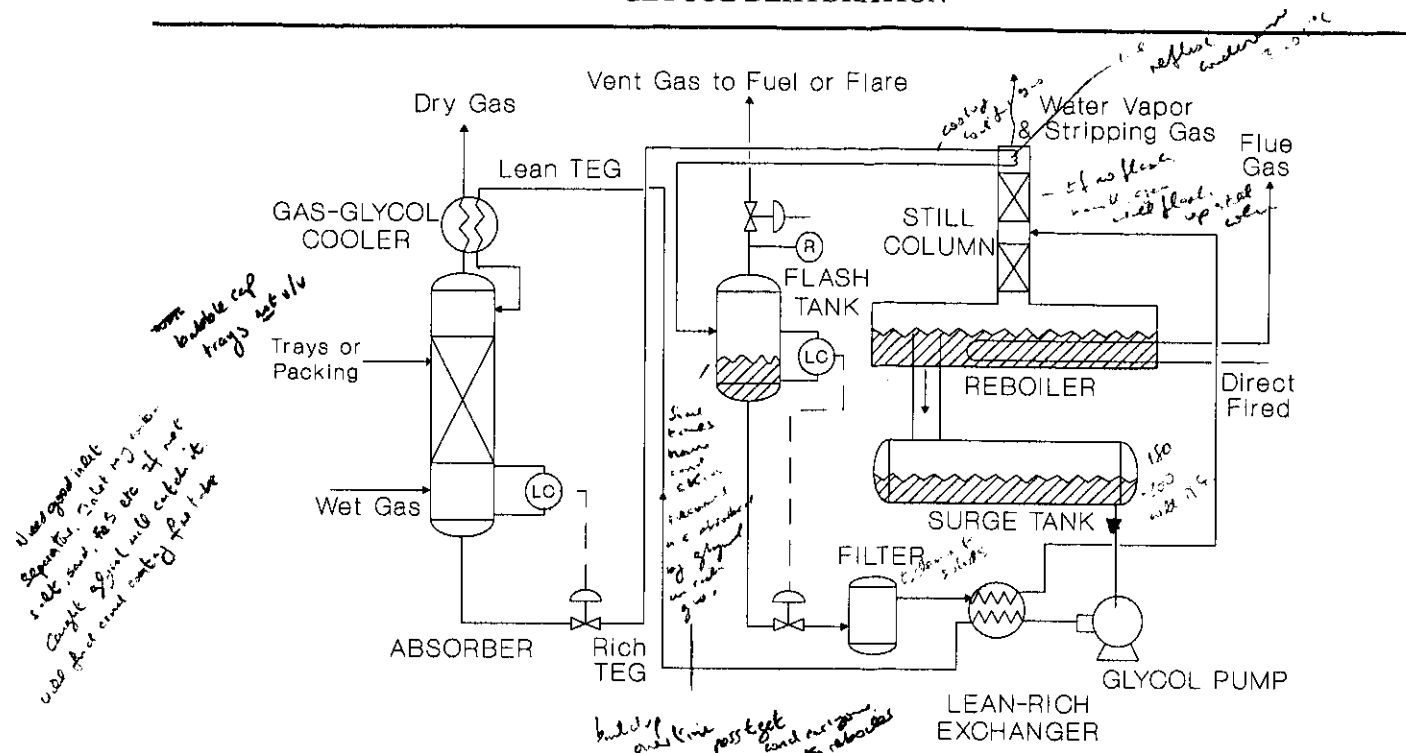


Figure 18.1 Basic Glycol Dehydration Unit

It is convenient to use the word "rich" to describe the bottom of the absorber and the word "lean" for the top. At the bottom, both the entering gas and glycol leaving are rich in water; at the top end they both are lean in water.

The rich glycol leaves the bottom of the absorber and flows to a reflux condenser at the top of the still column. The rich glycol then enters a flash tank where most of the volatile components (entrained and soluble) are vaporized. Flash tank pressures are typically 300-700 kPa [44-102 psia]. Leaving the flash tank the rich glycol flows through the glycol filters and the rich-lean exchanger where it exchanges heat with the hot lean glycol. The rich glycol then enters the still column where the water is removed by distillation.

The still column and reboiler are often called the regenerator or reconcentrator. This is where the glycol concentration is increased to the lean glycol requirement.

The regeneration unit shown is designed to operate at prevailing atmospheric pressure. The initial thermal decomposition temperatures of the glycols are

Glycol	Temperature
EG	165°C, 329°F
DEG	164°C, 328°F
TEG	206°C, 404°F
TREG	238°C, 460°F

rather - hot climates

These are the temperatures at which measurable decomposition begins to occur in the presence of air. DEG is no more stable than EG because it pyrolyzes in contact with carbon steel.

In the normal unit containing no air (oxygen), it has been found that one can operate the reboiler very close to the above temperatures without noticeable decomposition. So, they fix the composition of the lean glycol which leaves at its bubblepoint.

At the pressures involved, Raoult's Law applies:

$$x_w = \left(\frac{P}{P_v} \right) (y_w) \quad (18.1)$$

Where: x_w = mol fr water in lean glycol
 P = system pressure
 P_v = water vapor pressure at reboiler temperature
 y_w = mol fr water in the reboiler vapor (in equilibrium with x_w)

Once one has used the highest allowable temperature (to maximize vapor pressure, P_v), " P " and/or " y " must be lowered in value if lower water concentrations are needed. A vacuum pump or ejector may be used to lower pressure.

The addition of some other vapor to the reboiler will decrease the " y " for water. As shown in Figure 18.2(a), this may be accomplished with stripping gas. Any inert gas is suitable. A part of the gas being dehydrated, or exhaust from a gas-powered glycol pump (if used), is suitable. The quantity required is small. In theory, adding gas to a packed unit between the reboiler and surge tank is superior and will result in lower stripping gas rates. In commercial units it makes little difference how you get the gas into the reboiler so long as the quantity is right. It is common to use a distributor pipe along the bottom of the reboiler.

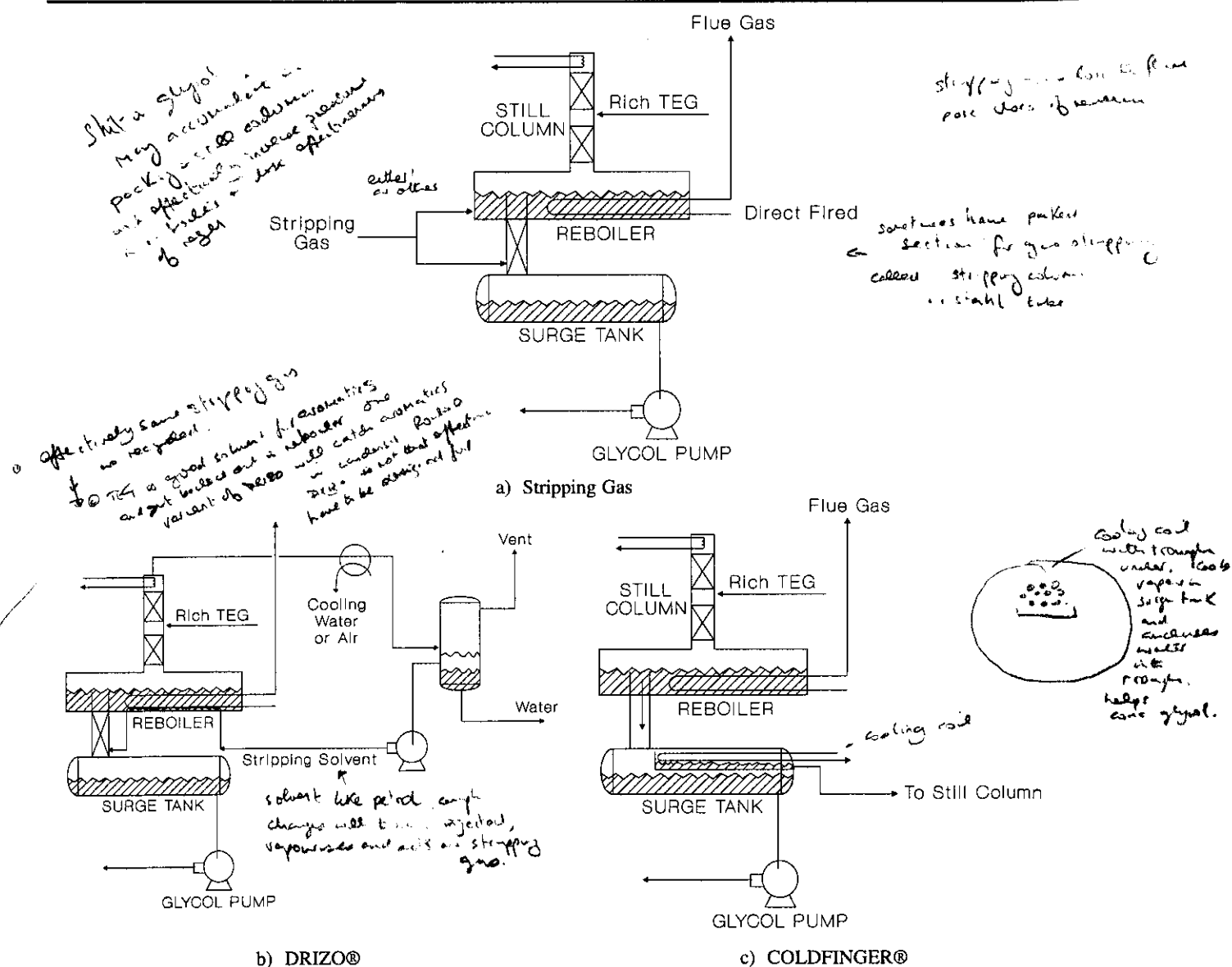


Figure 18.2 TEG Regeneration Alternatives

A second stripping gas alternative is close the stripping gas loop and use a material like iso-octane to the unit, as shown in Figure 18.2(b). It vaporizes at reboiler temperature but can be condensed and separated from the water in a 3-phase separator. The stripping solvent is then pumped back to the regenerator to complete the stripping loop. Sold under the trade name DRIZO®, this unit has the advantage of providing very high stripping gas rates with little or no venting of hydrocarbons. Glycol concentrations in excess of 99.99 wt% have been achieved with the DRIZO process. It has an added advantage of condensing and recovering aromatic hydrocarbons from the still column overhead. In fact, these units often operate with a stripping solvent which is not iso-octane but a mixture of aromatic, naphthenic and paraffin hydrocarbons in the C₅-C₈ range.

Figure 18.2(c) shows a third regenerator alternative called a COLDFINGER®. The COLDFINGER process achieves glycol enrichment by passing rich TEG through a cool "finger" inserted in the surge tank vapor space. This condenses a water-TEG mixture which is very rich in water. This mixture is drawn out of the surge tank by means of a trough below the "coldfinger" and is recycled back to the regenerator. The H₂O partial pressure in the vapor space is thus lowered and the lean glycol concentration increased. Lean TEG concentrations of 99.9 wt% have been achieved in COLDFINGER units without the use of stripping gas.

The unit shown in Figure 18.1 is typical. Figure 18.3 shows examples of two systems using TEG that incorporate additional features.

The upper flow sheet is for an offshore unit. The inlet scrubber is in the bottom of the absorber. Three-phase separation is required. The gas rises through a "chimney tray" to the absorber. The hydrocarbon and water are separated as shown. Three-phase separation saves on deck space and is less expensive, but many of the existing units are unsatisfactory because they provide inadequate separation.

The rich TEG from the chimney tray goes to a degassing pot (flash tank) which is operated at a high enough pressure to send the gas to fuel. In some systems the pressure is sufficient merely to enter the main flare system. The purpose of this is several fold: (1) use or dispose safely any volatile components picked up by the TEG in the absorber, and (2) minimize the presence of corrosive sulfur compounds and carbon dioxide in the high temperature reboiler.

The true solubility of paraffin hydrocarbons is very low in the glycols. But, separator carryover and entrainment does introduce hydrocarbons into the rich glycol. Many of these are "heavier" than air and can be a safety problem unless disposed of properly. In addition, aromatic components are very soluble in TEG. These can also be a safety concern when discharged to atmosphere at the top of the still column.

Both sulfur compounds and carbon dioxide are very soluble in water and react to some degree with the glycols. The degassing in the flash tank prior to the stripping column reduces their concentration and minimizes high temperature corrosion. This degassing is more efficient if the rich glycol is preheated first, as shown in the lower flow sheet in Figure 18.3.

External gas-TEG exchange is shown in the upper flow sheet, as opposed to external cooling in the lower one. Offshore it is suitable to use sea water cooling in plate exchangers, provided treated sea water is available for other purposes. In temperate latitudes aerial cooling also is a viable alternative.

An external glycol-glycol exchanger of the type shown in the lower flow sheet would be preferred if fuel is expensive. The closest economic temperature approach possible reduces reboiler heat load. Plate exchangers are widely used in this service.

Both flow sheets show a fired reboiler. The use of hot oil, steam, waste heat or electrical resistance coils are all suitable if they are readily available at the site. Oftentimes the use of electrical resistance offshore is cost-effective and safe.

No filter is shown on the upper flow sheet. It is on the low pressure side of the flash tank in the lower one. Location, pressure-wise, obviously affects cost. I prefer locating the filter at some point ahead of the

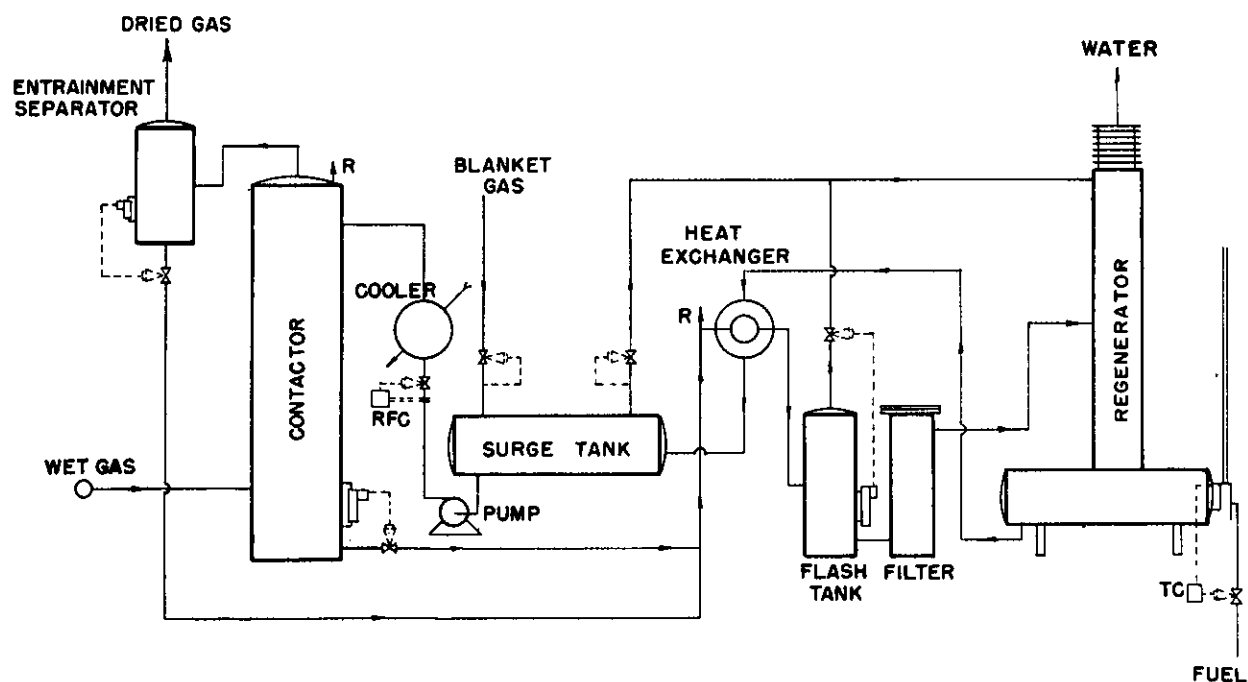
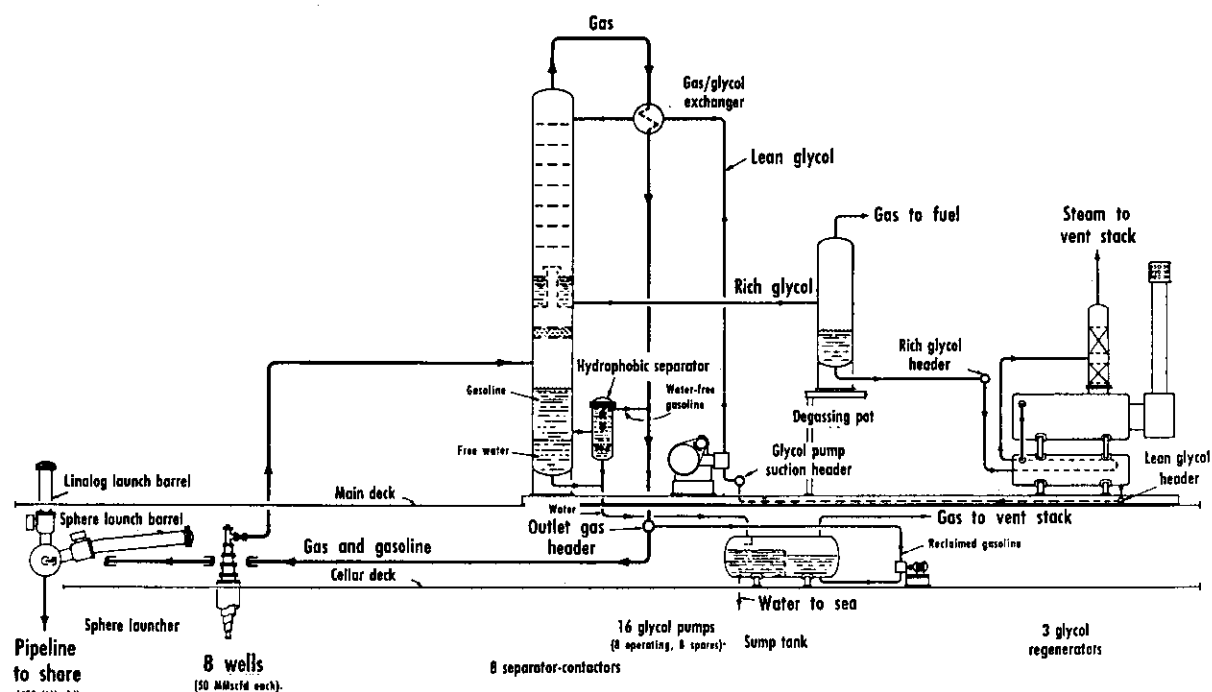


Figure 18.3 Flow Sheets for Two Different TEG Dehydration Systems

reboiler to minimize the "gunk" accumulating therein. For effective operation it is imperative that full-flow, glycol filters be installed in the system.

BASIC PROCESS DESIGN FACTORS

All factors controlling the behavior of absorption systems also apply for TEG dehydration. In fact, from a process viewpoint, TEG is one of the simpler absorption processes being employed in the petroleum industry.

In order to properly design a unit one needs to know maximum gas flow rate, maximum temperature and pressure, gas composition and required water dewpoint or content of the outlet gas. From these one can calculate:

1. The *minimum concentration* of TEG in the lean solution entering the top of the absorber required to meet outlet gas water specification.
2. The lean TEG circulation rate required to pick up from the gas needed amount of water necessary to meet the outlet gas water content specification.
3. The amount of absorber contact required to produce the necessary approach to equilibrium required in (1) above at the chosen circulation rate.

To obtain these answers it is necessary to have a vapor-liquid equilibrium correlation for a TEG-water system. From this basic input, one can size equipment and develop mechanical specifications.

The procedure that follows is very straightforward and can be performed manually. In all but a few exceptional applications, it will give results as reliable as more complex (appearing) methods. Following an outline of the basic calculation procedure, each major equipment component will be reviewed.

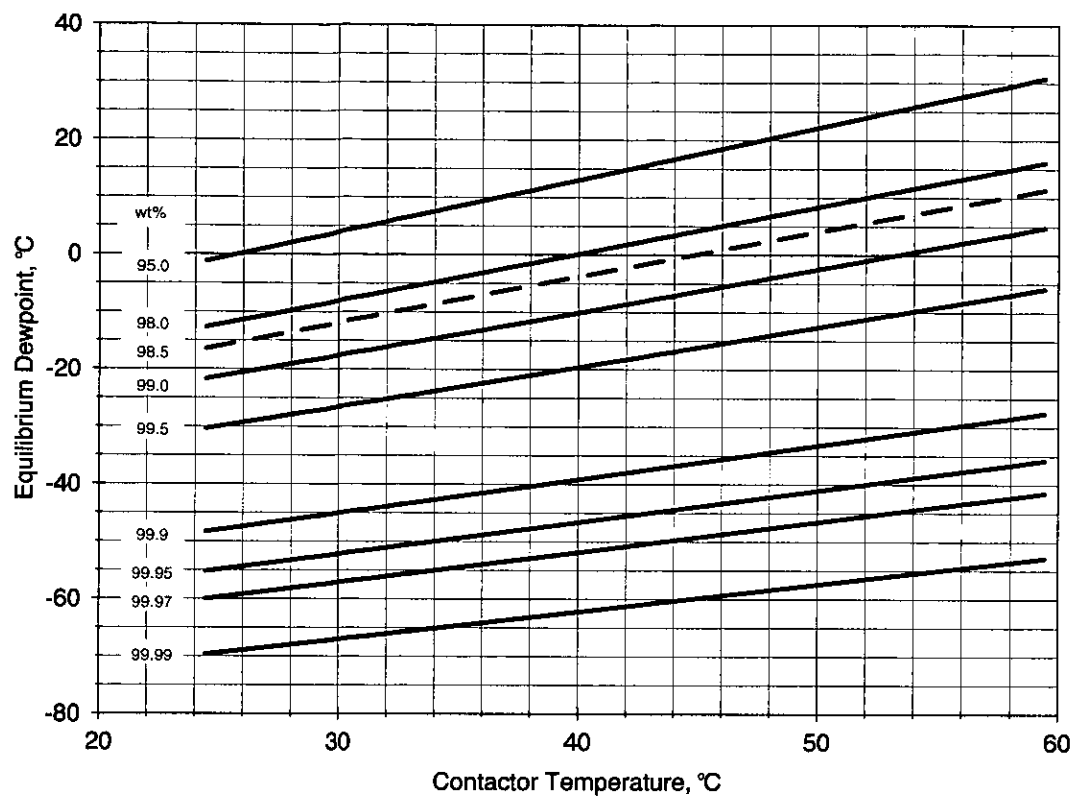
MINIMUM LEAN TEG CONCENTRATION

If water-saturated gas is placed in a static cell with a given concentration of TEG-water solution at a fixed P and T, equilibrium would be attained in time. Assuming the liquid had a sufficiently lower water concentration, water would transfer to this liquid from the gas. At equilibrium, the mol fraction water in the gas divided by its mol fraction in the liquid equals the K value for this system.

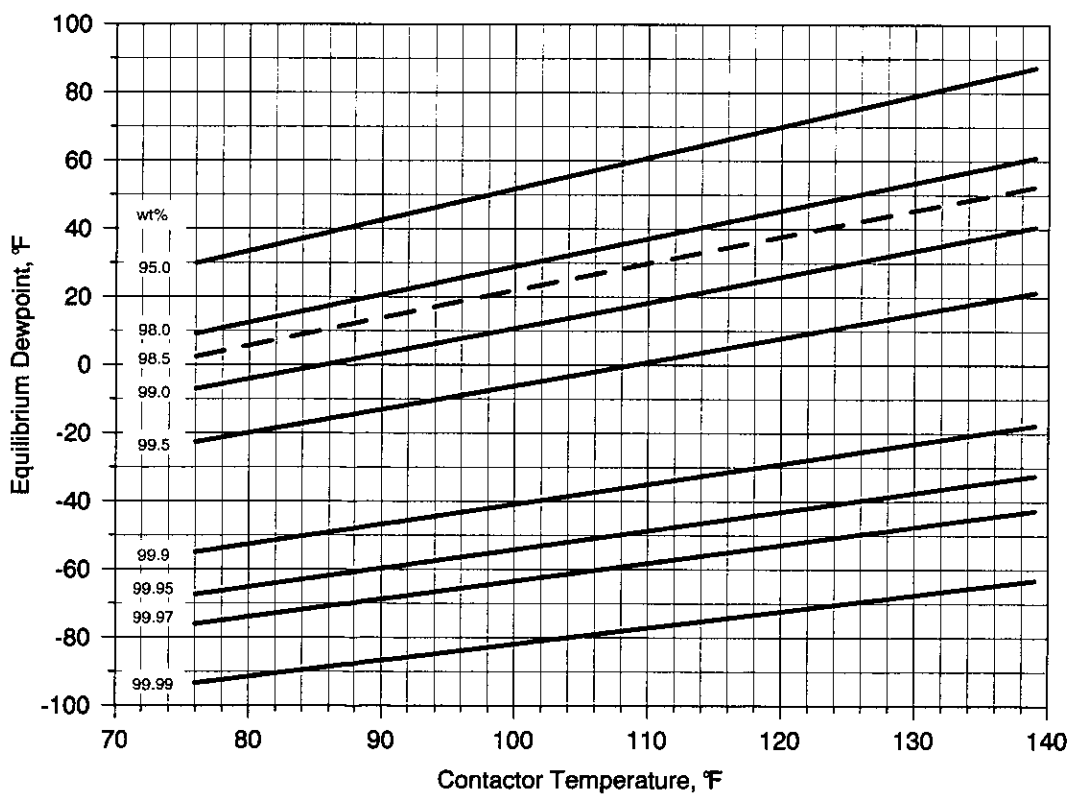
Figures 18.4(a) and 18.4(b) are based on equilibrium data published by Parrish, et. al.^(18.7) Several equilibrium correlations^(18.1,18.3-18.5,18.7) have been presented since 1950. Previous editions of this book presented an equilibrium correlation based on the work of Worley^(18.1). In general, the correlations of Worley^(18.1), Rosman^(18.5) and Parrish^(18.7) agree reasonably well and are adequate for most TEG system designs. All are limited by the ability to measure accurately the equilibrium concentration of water in the vapor phase above TEG solutions. The Parrish correlation has been included in this edition because equilibrium water concentrations in the vapor phase were determined at infinite dilution (essentially 100% TEG). The other correlations use extrapolations of data at lower concentrations to estimate equilibrium in the infinite dilution region. The effect of pressure on TEG-water equilibrium is small up to about 13800 kPa [2000 psia]^(18.4).

A TEG absorber is essentially isothermal. The heat of solution is about 21 kJ/kg [91 Btu/lbm] of water absorbed in addition to the latent heat. But, the mass of water absorbed plus the mass of TEG circulated is trivial to the mass of gas. So, the inlet gas temperature controls. The temperature rise due to heat of absorption seldom exceeds 2°C [4°F] except when dehydrating at pressures below about 1000 kPa [145 psia]. In low pressure service some temperature adjustment may be desirable.

The diagonal lines represent weight % TEG in a TEG-water mixture entering the top of the absorber. What is the lowest water dewpoint one could attain with a given concentration at a given temperature?



(a)



(b)

about 5
conc. no glycol
can be removed
without
stripping

Based on
- 10% of benzene
- 5% of water
- mixture of TEG
- Solubility of benzene
- 6% in 24 unit
of TEG
at 7 MPa (approx)
- 1% of water
- 10% of TEG

Figure 18.4 Equilibrium H₂O Dewpoint vs. Temperature at Various TEG Concentrations

Example 18.1: What equilibrium water dewpoint could be obtained at 40°C with a lean glycol solution containing 99.5 wt % TEG?

In Figure 18.4 locate 40°C on the abscissa, go vertically to the 99.5 wt % line and then horizontally to the ordinate. Read -19°C.

This water dewpoint could be attained in a test cell but not in a real absorber. The gas and TEG are not in contact for a long enough time to reach equilibrium. Numerous tests show that a well designed, properly operated unit will have an actual water dewpoint 5.5-8.5°C [10-15°F] higher than the equilibrium dewpoint. This "approach" to equilibrium can be used to specify minimum lean glycol concentration. The procedure is as follows.

1. Establish the desired outlet water dewpoint needed from sales contract specifications or from minimum system temperature.
2. Subtract the approach from (1) to find the corresponding equilibrium water dewpoint.
3. Enter the value in (2) on the ordinate of Figure 18.4 and draw a horizontal line.
4. Draw a vertical line from the inlet gas temperature on the abscissa.
5. The intersection of the lines in Steps (3) and (4) establishes minimum lean TEG concentration required to obtain the water dewpoint in Step (1).

If water content is specified or calculated in mass per unit gas volume, a water content, pressure, dewpoint temperature correlation is required. Note that the equilibrium water dewpoints on the ordinate of Figure 18.4 are based on the assumption the condensed water phase is a metastable liquid. At low dewpoints the true condensed phase will be a hydrate. The equilibrium dewpoint temperature above a hydrate is higher than that above a metastable liquid. Therefore, Figure 18.4 may predict dewpoints which are colder than those which can actually be achieved. The difference is a function of temperature, pressure and gas composition but can be as much as 8-12°C [15-20°F]. When dehydrating to very low dewpoints, such as those required upstream of a refrigeration process, the TEG concentration must be sufficient to dry the gas to the hydrate dewpoint.

Example 18.2: The gas sales contract specifies an outlet water content of 100 kg/10⁶ std m³ at a pressure of 6.9 MPa. The inlet gas temperature is 40°C. What minimum lean TEG concentration is required?

Metric: For 100 kg/10⁶ std m³ and 6.9 MPa, the equivalent dewpoint from a correlation is -2°C. If we use an 8°C approach the equilibrium dewpoint is -10°C. From Figure 18.4 at -10°C and 40°C contact temperature, wt % TEG = 99.0.

English: At 104°F [40°C] and 1000 psia [6.9 MPa] the dewpoint is 28°F for a water content of 6 lb/MMscf. An approach of 14.4°F [8°C] gives an equilibrium temperature of about 14°F. From Figure 18.4(a), lean TEG concentration equals 99.0 wt %.

The dashed line in Figure 18.4 at about 98.5 wt% represents the concentration of lean TEG that can be produced routinely in a regenerator operating at standard atmospheric pressure and 204°C [400°F]. This is a safe value for design and specification purposes. Concentrations of 98.7-98.8 wt% are common; some to 99.1 wt% have been reported but represent a special case where incoming hydrocarbons provided natural stripping and/or the pressure was lower than standard atmospheric.

Since the capital cost of ordinary gas stripping accessories is trivial, they always should be included. Conditions can change to where they may be required.

It is necessary to fix a lean TEG concentration for subsequent calculations. For the first consideration, use the results from Figure 18.4. If the concentration obtained is less than 98.5 wt%, use 98.5 wt% for the calculation unless you plan to reduce the reboiler temperature below 204°C [400°F].

The minimum lean TEG concentration may not be the one used. A higher concentration than this may be specified to minimize circulation rate and optimize cost.

TEG REGENERATION

A given lean TEG concentration is produced in the reboiler and still column (regenerator) section by control of reboiler temperature, pressure and the possible use of a stripping gas. So long as no stripping gas is used, the concentration of the lean TEG leaving the reboiler is independent of the rich TEG entering.

When stripping gas is used, the concentration of rich TEG leaving the absorber is found by a water material balance around that absorber. By definition

$$\text{wt \% rich TEG} = \frac{\text{wt lean TEG}}{\text{wt lean TEG} + \text{wt water absorbed} + \text{wt water in lean TEG}} (100) \quad (18.2)$$

The weight quantities in this equation may be found per unit of time or per unit of gas flow. In any case, the values used depend on circulation rate. This rate depends on dewpoint requirements, lean TEG concentration, amount of absorber contact and economics. The latter dictates a rather low circulation rate. This rate usually will be 16-40 liters of lean TEG solution per kg of water absorbed from the gas [2-5 U.S. gal/lb water]. The minimum rate is governed by the rate required for effective gas-liquid contact in the absorber; the maximum is limited by economics.

Because this regeneration takes place at low pressure, under ideal gas conditions, the calculation is very routine. Figure 18.5 has been calculated to predict regenerator performance.^(18.2)

The minimum wt% lean TEG on the top abscissa is found from Figure 18.4. The wt% of rich TEG on the bottom abscissa is found from Equation 18.2. The rich TEG concentration can also be determined from Equation 18.3.

$$\text{Rich TEG} = \frac{(\rho) (\text{Lean TEG})}{\rho + (1/m)} \quad (18.3)$$

		Metric	English
Where:	ρ = liquid density	1.12 kg/L	9.3 lb/U.S. gal
	m = lean TEG rate <i>Litres TEG / Kg water needed</i>	L/kg	U.S. gal/lb
	Rich TEG = wt% TEG in rich TEG solution		
	Lean TEG = wt% TEG in lean TEG solution		

5

The diagonal lines in the lower left portion of Figure 18.4 represent various amounts of stripping gas. It is seldom that one would use more than 0.08 m³ gas per liter of TEG [10 scf/U.S. gal]. At some point the practical impact of stripping gas diminishes with rate.

Three temperature lines are shown. Where high concentrations are desired, the specification of 204°C [400°F] is normal unless the gas being dehydrated contains oxygen. This is close to the thermal decomposition temperature (in air). In the usual case where the natural gas is oxygen free, the use of 204°C [400°F] has proven very satisfactory.

The normal reboiler temperature is 204°C. If no stripping gas is used, rich glycol composition has no effect on performance. If no vacuum is to be employed, proceed vertically from the temperature line to the lean glycol composition.

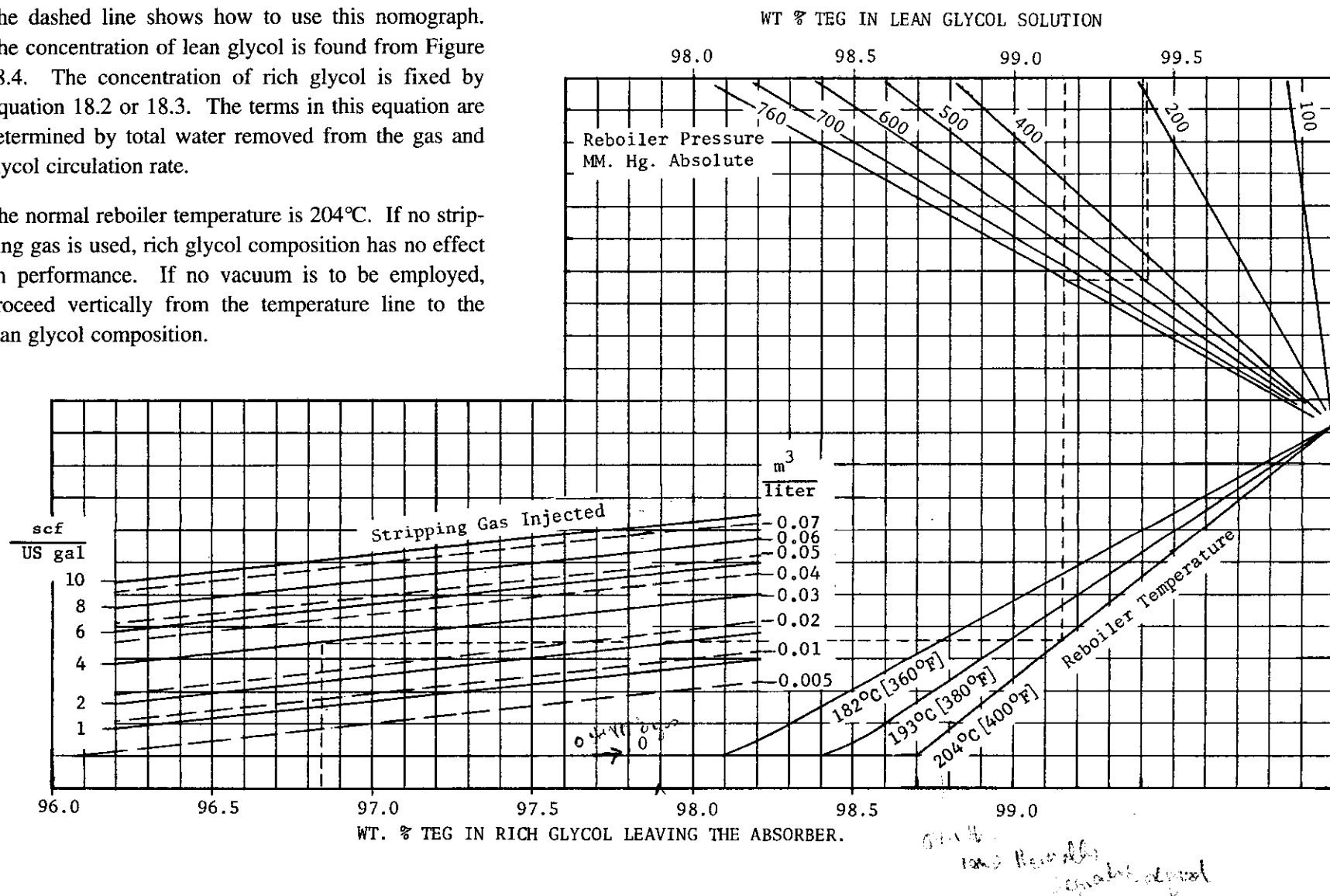


Figure 18.5 Nomograph for Estimating Regenerator Performance as a Function of Pressure, Reboiler Temperature and Stripping Gas

The diagonal lines at upper right in Figure 18.5 represent the effect of regeneration pressure in mm Hg,

$$760 \text{ mm Hg} = 14.7 \text{ psia} = 101.325 \text{ kPa}$$

$$100 \text{ kPa} = 1 \text{ bar} = 750 \text{ mm Hg}$$

Unless a vacuum is being used, it is customary to use the 760 mm Hg line for design calculations.

Notice that at 760 mm Hg pressure and a reboiler temperature of 204°C, Figure 18.5 shows a lean TEG concentration of 98.7 wt%. If in using Figure 18.4 you obtain a concentration less than this, use 98.7 wt% as the desired concentration when utilizing Figure 18.5.

The general procedure for using Figure 18.5 is as follows:

1. Atmospheric Pressure, No Stripping Gas -

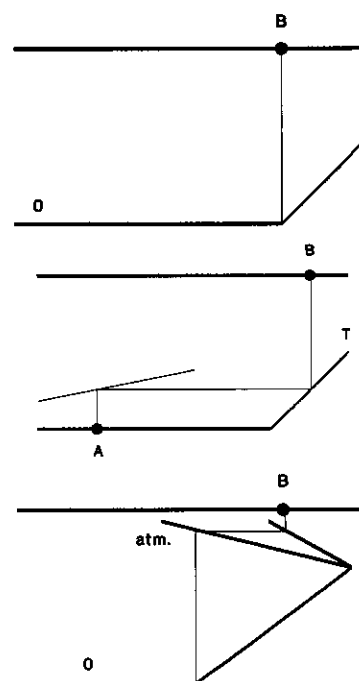
Wt% rich glycol is not a variable. Proceed vertically from 0 stripping gas and temperature line intersection. You will read 98.7 wt% TEG at 204°C; 98.4 wt% at 193°C.

2. Atmospheric Pressure, Stripping Gas -

- a. Proceed vertically from B to temperature line and then horizontally.
- b. Proceed vertically from A.
- c. Intersection of two lines from A and B fixes amount of stripping gas.

3. Vacuum, No Stripping Gas -

- a. Proceed vertically from intersection of 0 gas line and temperature line to atmospheric line (760 mm Hg).
- b. Proceed horizontally from point in (a) to pressure line necessary to fix value of point B.



In that rare case where both stripping gas and vacuum are used, procedures (2) and (3) are combined.

Example 18.3: An example is shown on Figure 18.5 for use of stripping gas and vacuum. A 96.84 wt% rich glycol enters a regenerator using 0.03 m³ of stripping gas per liter of glycol solution [4 scf/U.S. gal]. Proceeding to 204°C and then vertically, one reads 99.16 wt% if atmospheric pressure is used. If a vacuum is employed and the absolute pressure is 500 mm Hg, the lean glycol concentration is 99.41 wt%.

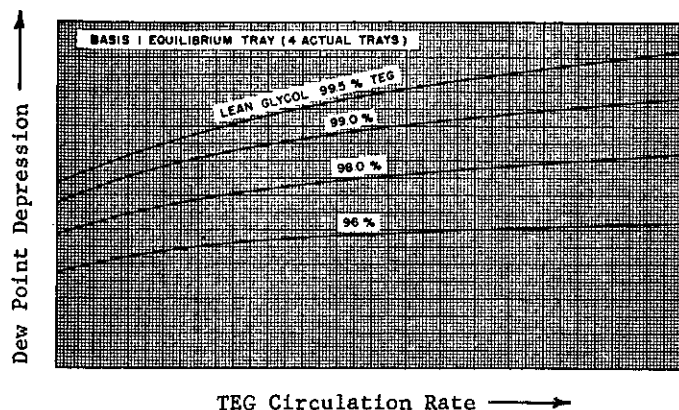
As a general rule, vacuum is avoided unless necessary to simplify unit operation. Vacuum pumps can be a nuisance. An ejector can be used to produce necessary vacuum in the right circumstances.

Figure 18.4 is based on 1 equilibrium stage in the regenerator. Most regenerators will contain more than 1 equilibrium stage, particularly if a stripping column is installed between the reboiler and surge tank. For this reason, actual stripping gas rates will typically be less than those predicted from Figure 18.4.

CIRCULATION RATE – ABSORBER CONTACTS

The above calculation is dependent on glycol circulation rate if stripping gas or vacuum are used. Although a number of circulation rates are possible, the minimum feasible one should be used. As circulation

rate increases, so does operating cost. The minimum feasible rate is fixed by absorber characteristics and cost.



The figure at left shows the effect of TEG concentration and circulation rate on dewpoint depression for a fixed amount of absorber contact of absorber contact. Notice that the curves become relatively flat at high circulation rates.

For ordinary pipeline water dewpoint control, the glycol circulation rate usually will be 16-40 liters per kg water absorbed from the gas [2-5 U.S. gal/lb water]. A circulation rate above this range can be justified economically only in special applications for it results in excess utility consumption.

For a given circulation rate a given number of absorber contacts are needed. The relationship between rate and amount of contact is very adequately described by the Kremser-Brown method described in Chapter 17. In the glycol application, mol fractions may be used in instead of regular absorption parameters because of the concentrations involved. The basic absorption equation thus may be rewritten as:

$$E_a = \frac{y_{N+1} - y_1}{y_{N+1} - y_0} = \frac{A^{N+1} - A}{A^{N+1} - 1} \quad (18.4)$$

- Where:
- y_{N+1} = mol fr water in entering wet gas
 - y_1 = actual mol fr water in dried gas leaving
 - y_0 = water content of dried gas if it is in *equilibrium* with the entering lean glycol (value is less than y_1)
 - A = absorption factor, $A = L/(KV)$
 - L = glycol circulation rate, moles/unit time
 - V = gas flow rate, moles/unit time
 - K = equilibrium constant for water between water in gas and water in a TEG-water solution, $y = Kx$
 - N = no. of theoretical plates in the absorber

The mol fraction water, y_w , is related to W , the mass of water per standard volume of gas by a fixed conversion factor. As noted in Chapter 6,

$$y_w = 1.33(E-06) W$$

Where: $W = \text{kg}/106 \text{ std m}^3$

$$y_w = 2.11(E-05) W$$

Where: $W = \text{lbm}/\text{MMscf}$

Thus,

$$E_a = \frac{y_{N+1} - y_1}{y_{N+1} - y_0} = \frac{W_{N+1} - W_1}{W_{N+1} - W_0} \quad (18.5)$$

The subscripts on "W" have the same significance as on "y."

For a given calculation the values of inlet water content, outlet water content, gas flow rate, and absorber pressure and temperature are fixed. Using a correlation for determining the equilibrium K value of water in a TEG-water system, values for y_0 (or W_0) and K are available. The only variables left in Equation 18.4 are L (the TEG circulation rate) and N (the number of theoretical trays). In theory, there are an infinite

number of combinations of L and N that satisfy Equation 18.4. In practice, the choices are limited by economics and absorber performance.

The cost of purchasing and operating any absorption unit is a function of circulation rate. It is thus good practice to operate at, or near, the minimum rate necessary to meet absorption specifications.

Figure 18.6 is a plot of Equation 18.4 that is convenient for manual calculations. This uses what could be called an *overall absorption factor*. The ratio L/V varies slightly throughout the absorber. L_0 is the rate of lean TEG entering the top tray and V_{N+1} is the gas rate entering the bottom tray. As a later numerical example will illustrate, the variation of L/V will have a calculable, but usually nonsignificant, effect on unit design.

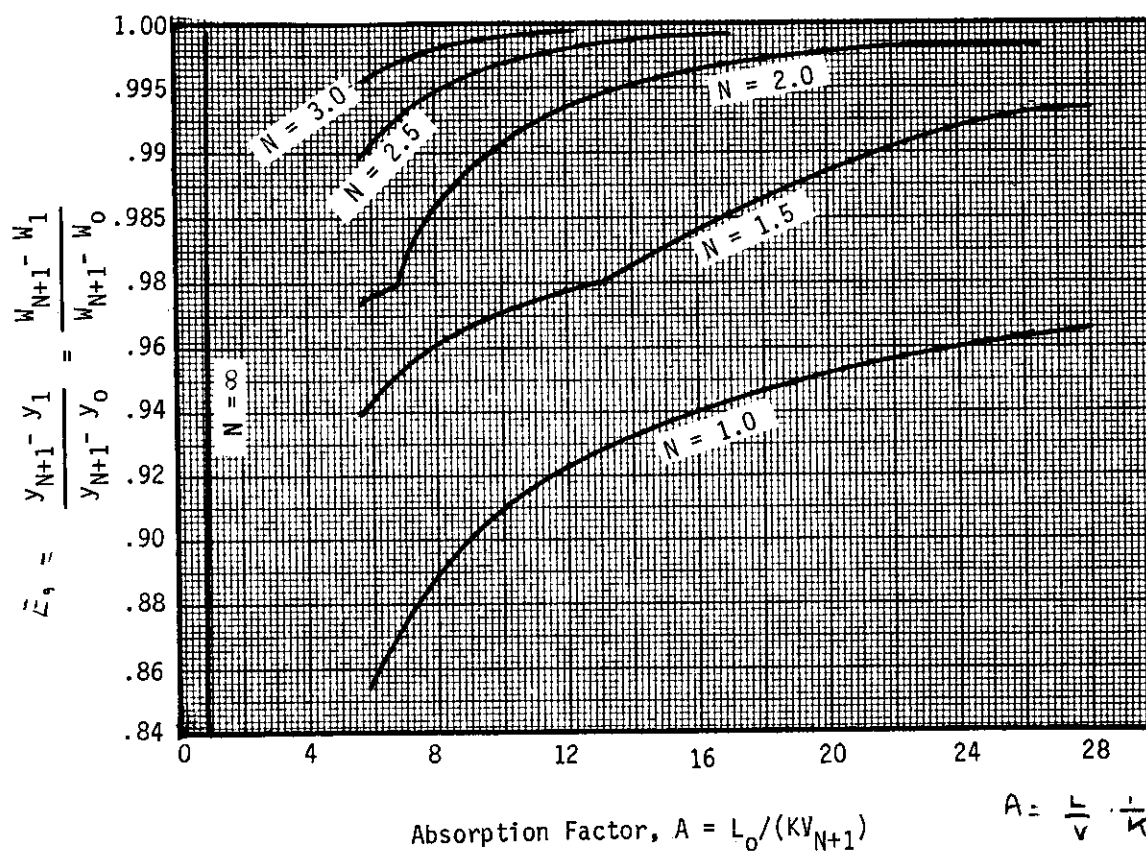


Figure 18.6 Plot of Kremser-Brown Equation

The left-hand ordinate of Figure 18.6 could be called *absorption efficiency*. It is the actual amount of water removed, divided by the maximum amount removable. The values of N encompass the range of theoretical trays usually employed. The line for N equals infinity represents the minimum absorption factor, e.g., minimum circulation rate. All curves for finite values of N become asymptotic to this line. Note that the scale changes on the ordinate at 0.98.

Calculation of Lean TEG Rate for a Given Absorption Efficiency and N -

1. Calculate y_0 (or W_0)
2. Determine absorption efficiency
3. Use Equation 18.4 or Figure 18.6 to find absorption factor A for a given value of N
4. Knowing V_{N+1} and K , solve A for L_0 , the lean TEG circulation rate

Calculation of N for a Given Lean TEG Rate and Absorption Efficiency -

1. Calculate y_0 (or W_0)
2. Determine absorption efficiency
3. Calculate absorption factor A
4. Determine N from Equation 18.4 or Figure 18.6

It is usual to repeat the calculation to obtain three lean glycols rate/absorber contact values that satisfy the required absorption efficiency. The final choice is economic. This usually involves selection of standard modules to make up the unit.

This calculation should be made at the lowest pressure and highest temperature anticipated for the entering wet gas, to obtain the maximum water loading. Historically, the tendency has been to choose a design temperature lower than that actually obtained.

The overall tray efficiency in a well-designed TEG unit will vary from 25-40%. It is recommended that 25% be used for most applications. This provides an affordable safety factor to help compensate for the inherent errors in the design specifications.

Equilibrium Relationships

Various studies have been made of the equilibrium behavior of water in the TEG-water system.^(18.3-7) All provide rather consistent data. The use of an activity coefficient (γ) is a convenient and reliable method for calculating K. Using this relationship

$$K = (y_w)(\gamma) = (B)(W)(\gamma) \quad (18.6)$$

- Where:
- K = equilibrium constant for water in a TEG-water system
 - y_w = mol fr water in the gas at saturation over 100% liquid water (from regular water content correlation)
 - γ = activity coefficient for water in the TEG-water system as found from Figure 18.7
 - W = water content on a mass per volume basis, at saturation, as found from a regular water content correlation
 - B = 1.33(E-06) when W = kg/10⁶ std m³ (measured at 15°C and 100 kPa)
= 2.11(E-05) when W = lbm/MMscf (measured at 60°F and 14.7 psia)

Notice that γ and thus K vary with TEG concentration and temperature, which in turn varies throughout the absorber. An average K at average concentration cannot be found until the circulation rate is fixed. So, a simple trial-and-error calculation is involved. One can assume the inlet lean TEG concentration as a first try. For most dehydration applications the increase in K from the lean to rich TEG is roughly proportional to the increase in L/V, so the absorption factor ($A = LV/K$) remains relatively unchanged.

In the absorption efficiency term

$$y_0 = K x_0 \quad \text{and} \quad W_0 = (W)(\gamma)(x_0) \quad (18.7)$$

Where: x_0 = mol fr water in the lean TEG entering the absorber

This may be calculated from X_{gl} , the weight percent TEG in the lean solution entering the absorber. This must be not less than the minimum value required from Figure 18.4.

$$x_0 = \frac{(100 - X_{gl})/18}{[(100 - X_{gl})/18] + (X_{gl}/150)} \quad (18.8)$$

1.2 mols H₂O / mols TEG = mols H₂O / mols H₂O

Equation 18.8 is shown graphically in Figure 18.8.

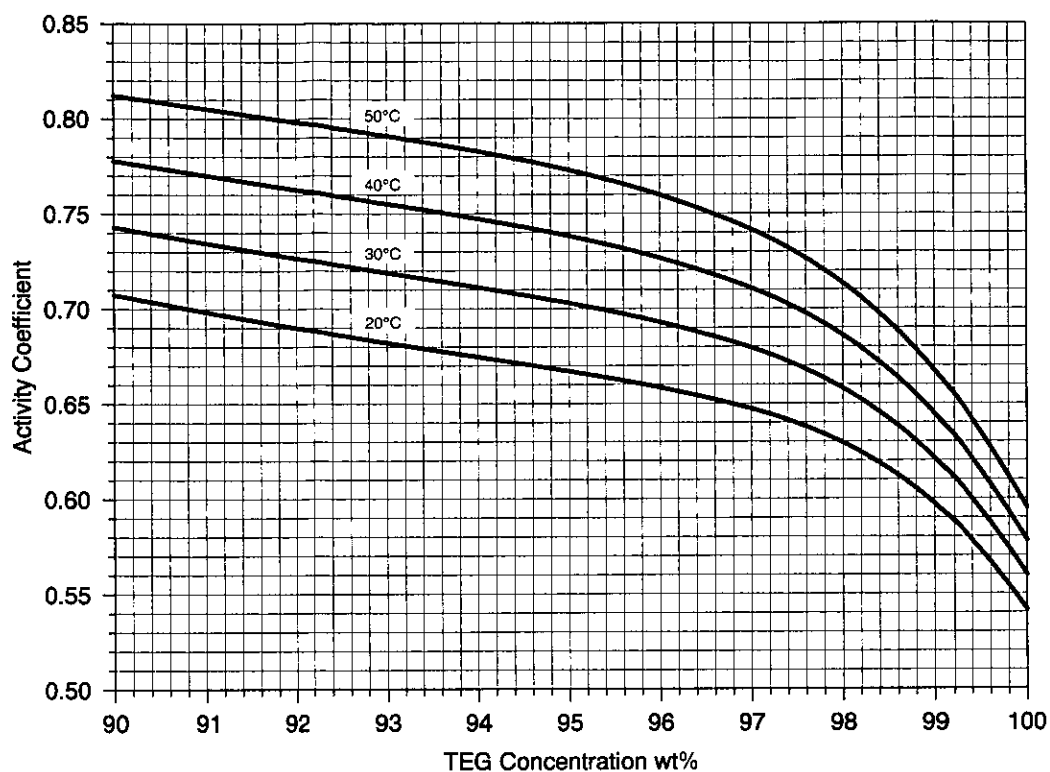


Figure 18.7 Activity Coefficient for H_2O Concentration at Various Temperatures

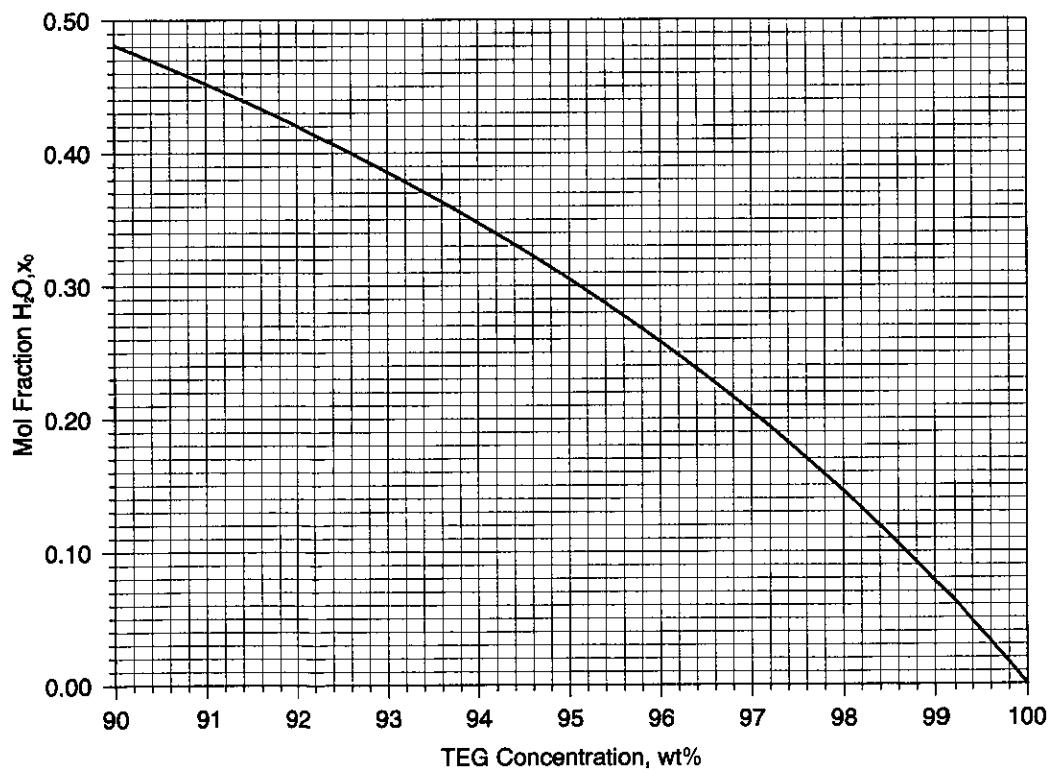


Figure 18.8 Equation 18.8 Mol Fraction H_2O vs. TEG Concentration

The above procedure using the Kremser-Brown approach to absorption is as good as a tray-to-tray balance around the absorber. As reviewed in Chapter 17, this is true because there is little gas shrinkage and A is essentially constant throughout the tower.

The use of other equilibrium K values will have little effect on contactor design. The required lean glycol concentrations may differ but the difference is normally less than the random error in process specifications and is statistically insignificant.

Example 18.4: Calculate the circulation rate of 98.7 wt% lean TEG needed to dry 10^6 std m^3/d of gas at 7.0 MPa and 40°C in a six tray absorber (1.5 theor. tray) to achieve an exit gas water content of $117 \text{ kg}/10^6 \text{ std } m^3$. The inlet water content is $1100 \text{ kg}/10^6 \text{ std } m^3$ (saturated gas).

1. From Equation 18.8, $x_0 = 0.099$
2. From Figure 18.7, $\gamma = 0.66$
3. W is the water content of saturated gas at 7.0 MPa and 40°C or $1100 \text{ kg}/10^6 \text{ std } m^3$ in this case.
4. From Equation 18.7, $W_0 = (1100)(0.66)(0.099) = 71.9 \text{ kg}/10^6 \text{ std } m^3$
5. The left-hand side of Figure 18.6 is: $(1100 - 117)/(1100 - 71.9) = 983/1048 = 0.956$
6. From Figure 18.6, for $N = 1.5$, $A = 7.3$

(In order to solve for L one must recognize that it is different at each point in the absorption tower. The conservative approach is to assume the gas volume is constant and solve for L_0 from A . This will yield a circulation rate sufficiently close to a more rigorous calculation.)

7. $L_0 = (A)(K)(V_{N+1})$

From Equation 18.6, $K = (1.33 \times 10^{-6})(1100)(0.66) = 0.000967$

$V = 1739 \text{ kmol/h}$

So, $L_0 = (7.3)(9.67 \times 10^{-4})(1739) = 12.3 \text{ kmol/hr}$

8. MW lean glycol = $(0.099)(18) + (0.901)(150) = 137$

9. $\text{kg TEG/h} = (12.3)(137) = 1685$

10. Density of TEG is $1.12 \text{ g/cm}^3 = 1.12 \text{ kg/liter}$

Circulation rate is $1681/1.12 = 1500 \text{ liter/h}$

In one hour $(1100 - 117)/24$ or $41.0 \text{ kg H}_2\text{O}$ is absorbed.

Circulation rate is $1500/41 = 36.6 \text{ liter/kg H}_2\text{O absorbed}$.

$$\frac{1 \times 10^{-4}}{23.66 \times 10^{-4}}$$

In traditional English units the calculation follows the same format.

- 1., 2. The same
3. $W = 67 \text{ lb/MMscf}$
4. $W_0 = (0.66)(67)(0.099) = 4.38 \text{ lb/MMscf}$
5. $(67 - 7)/(67 - 4.38) = 0.958$
6. $A = 7.5$
7. $K = (2.11 \times 10^{-5})(67)(0.66) = 0.000931$
 $V = (34.9)(110) = 3839 \text{ lb-mol/hr}$
 $L_0 = AKV_{N+1} = (7.5)(9.31 \times 10^{-4})(3839) = 26.8 \text{ lb-mol/hr}$
8. MW = 137
9. $\text{lb/TEG/hr} = (26.8)(137) = 3672$
10. Density of TEG is about 9.3 lb/U.S. gal
Circulation rate is $3672/9.33 = 394 \text{ U.S. gal/hr}$
In one hour a total of 92 lb of water is absorbed.
Circulation rate is $394/92 = 4.3 \text{ U.S. gal/lb water absorbed}$

Example 18.4 illustrates the calculation procedure outlined above and used in the accompanying computer program.

When using the Kremser-Brown method, the terms V and L must be expressed in molar units. For the glycol solution this requires estimation of the MW. The molecular weight of a TEG water solution may be calculated as follows:

$$MW = 18 x_0 + 150 (1 - x_0) \quad (18.9)$$

The actual circulation rate will be different from this because operating conditions always differ to some degree from those specified. This number will be used to select a pump size. It is then sound practice to choose other components to have a capacity compatible with that of the pump.

For the example case, would we buy a 1.5 theoretical tray absorber? Probably not! The circulation rate calculated is toward the high end of the economic range. A 7 or 8 actual tray (1.75–2.0 theoretical trays) might well be specified to provide valuable flexibility and inexpensive "insurance."

ABSORBER DESIGN

The absorber will be sized using the criteria outlined in Chapters 11 and 17. Special consideration must be given to the low circulation rate, glycol viscosity and its foaming tendencies.

A basic decision is whether to use some kind of tray or packed tower. With tray towers the choice is between valves and bubble caps. The former is more efficient at design capacity, but at lower flow rates glycol "weeping" may produce unsatisfactory water dewpoints. Bubble caps certainly are a safe choice in service where widely fluctuating gas flow rates are anticipated.

Regardless of the type of tray, a minimum spacing of 24 inches is recommended. It is essential that a stable foam not fill the gas space between trays to prevent excess glycol loss. This spacing also allows a suitable liquid level in the downcomers.

Tray hydraulics design is critical because of the low circulation rate. Liquid can by-pass caps or valves in some areas of the tray, ineffective gas-liquid contact can occur with low gas rates, and liquid levels can be unstable. In situations like this, absorber performance can vary markedly with gas and liquid rates. A higher than calculated liquid rate may be necessary to provide the tray efficiency required.

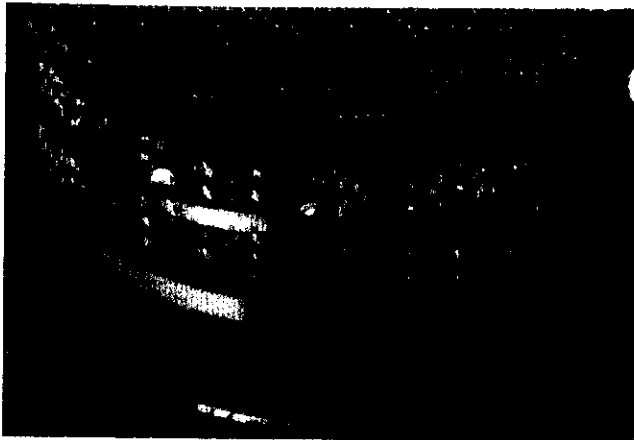
Remember that the minimum TEG concentration found from Figure 18.4 assumes that the tower provides the contact required, e.g., the proper number and efficiency of trays. The result desired is achievable with that concentration only if both the mechanical design of the absorber and the circulation rate are proper. On the other hand, if you don't have at least that concentration, the result is unattainable, regardless of the amount of contact or circulation rate.

Random packing has been used successfully for gas-liquid contact in a glycol absorber but is not generally recommended. One can encounter liquid distribution problems. If undue foaming occurs, the tower can flood at lower than normal gas rates and cause excess glycol losses. Solids in the gas tend to plug the packing.

The use of structured packings is standard practice in glycol dehydration service today. Some of the more common types include Mellapak, Montzpak and Flexipak and Gempak. Structured packings offer the advantage of considerably smaller diameter towers as compared to trayed contactors for the same gas rate. This feature can significantly reduce capital costs, especially offshore. In addition, glycol losses may be reduced due to less agitation of the glycol solution at the top of the absorber. Mass transfer is somewhat better than trays but the real savings is in tower diameter.

For sizing packed glycol contactors using structural packing, the following parameters are useful.

	Metric	English
K_s	0.1 m/s	0.33 ft/sec
HETP	1.5–2.0 m	60–80 in.

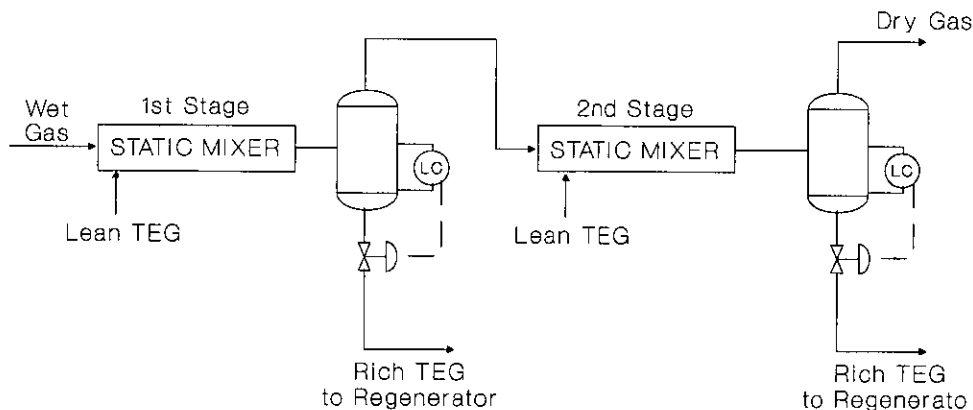


Courtesy Nutter Engineering

Structured packing consists of prefabricated elements approximately 0.2 m [8 in.] thick. Each section is oriented 60° or 90° relative to the adjacent unit to insure good liquid distribution. Sealing is provided at the vessel wall with wall wipers which come with the packing. Two sections of structured packing are shown to the left.

One of the primary applications of structured packing has been to increase the gas handling capacity of an existing trayed contactor. Retrofits increasing absorber capacity by 100% have been reported. When retrofitting a contactor adequate pump and regenerator capacity must also be checked.

Some vendors offer an absorber other than a conventional vertical unit. One design uses two or three cocurrent contactors in series separated by scrubbers. The cocurrent contactors are static mixers. This design often has a lower installed weight when compared to a conventional contactor, and may offer a cost advantage when a high dewpoint depression is not required.



Static mixers have also been used upstream of conventional contactors to add some portion of a theoretical stage to the unit.

REGENERATION AND HEAT EXCHANGE

The glycol circulated must be heated to reboiler temperature and then cooled before re-entering the absorber. Efficient rich glycol-lean glycol heat exchange will minimize fuel and/or utilities cost.

The detailed calculation involves a heat and material balance around the regeneration system for the maximum anticipated glycol circulation rate. The reboiler heat load depends on the efficiency of exchange.

The Reboiler

Commonly a direct-fired reboiler is used. The outside temperature of the fire tube surface, covered by glycol, is surprisingly cool, if the unit is properly fired. Figure 18.9 shows one temperature profile obtained on test.

Outside surface temperature may be lower than steam or hot oil units because of high film resistance on the low pressure gas side. This temperature is important to prevent cracking of any hydrocarbons present and degradation of the glycol.

To maintain proper skin temperature, prevent "hot spots" and obtain satisfactory fire tube life, it must have sufficient area. The following are recommended:

Max. heat flux across fire tube wall	25 kW/m ² , [8000 Btu/hr-ft ²]
Recommended heat flux for max. life	20 kW/m ² , [6000 Btu/hr-ft ²]
Burner capacity	30 kW/m ² , [10 000 Btu/hr-ft ²]

Notice that the burner possesses extra capacity to obtain firing flexibility.

The typical heat balance will indicate a reboiler heat load of 390–450 kJ/liter of TEG circulated [1400–1600 Btu/U.S. gal]. Experience has shown that extra heating capacity is desirable. Based on the maximum pump capacity, I recommend that the minimum reboiler rating be

$$560 \text{ kJ/liter TEG [2000 Btu/U.S. gal TEG]}$$

The direct fired reboiler is very efficient but there can be safety problems, particularly offshore. Hot liquid, steam, electrical resistance coils and waste heat can be used. The choice depends on what else is on the platform. The heat load of the glycol unit is rather small so it seldom justifies a separate source.

The Still Column

This is a packed column except in large sizes. The upper part of this is really a rectification section to prevent glycol loss. No formal amount of reflux is necessary with TEG. Just enough is needed to keep the packing wet. Since the vapor load is low, the sizing of this column is not critical. It should, however, conform generally to the diameter approximated in Equation 18.10.

$$d = (A)(m)^{0.5} \quad (18.10)$$

Where:

- d = diameter of packed tower
- m = glycol circulation rate
- A = empirical constant

Metric	English
cm	in.
L/min	U.S. gal/min
11.9	9.1

Glycol–Glycol (Lean-Rich) Heat Exchanger

This is the basic heat exchanger. Its efficiency has a direct effect on reboiler heat load. The rich glycol from the absorber enters at about the temperature of the inlet gas (Pt. 1). The lean glycol from the regenerator (Pt. 3) enters usually at about 204°C [400°F]. The temperature of the lean glycol at Pt 4 should not be greater than 60–65°C [140–149°F].

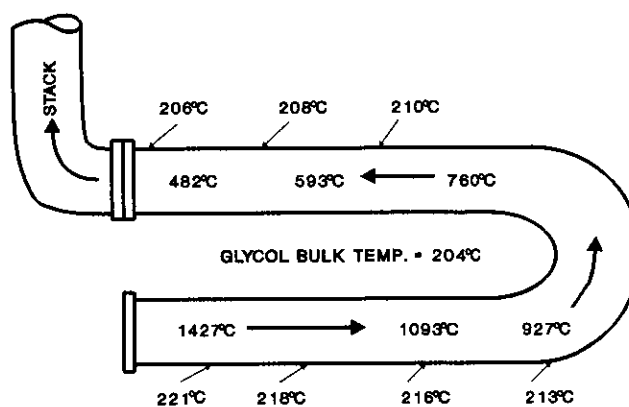
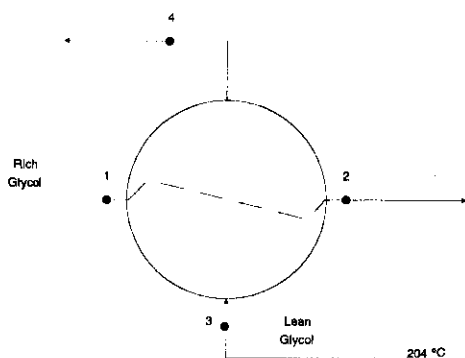


Figure 18.9 Typical Temperature Profile in Gas Fired Glycol Reboiler



In most cases a 15-20°C approach in the heat exchanger is desirable. If it is too high the reboiler and glycol cooler duties will increase. In this exchanger

$$Q_{3-4} = Q_{1-2} = m_1 (h_2 - h_1) = m_3 (h_3 - h_4) \quad (18.11)$$

The easiest way to make this enthalpy calculation is to look up the average specific heat of the glycol in Appendix 18A and multiply it by ΔT across the heat exchanger to find Δh for the lean mixture. The temperature of the rich glycol entering the heat exchanger will be about the same as that of the entering wet gas.

If an efficient glycol-glycol exchanger is used there may be no need for an additional cooler on the lean glycol stream. A cooling coil in the top of the absorber is not recommended for any application other than small wellhead units. The most common types of glycol-glycol exchangers are pipe-in-pipe and plate. The first are used on small units, while the latter find extensive use in larger units and offshore. Plate exchangers provide effective heat exchange but are very susceptible to fouling in dirty service. Clean filtered glycol is imperative if plate exchangers are used.

FILTERS

Good filtration is critical. The full-flow type is preferred. I recommend two filters in parallel, with no by-pass lines, so that full filtration is assured.

A cloth fabric element that is capable of reducing solids to about 100 ppm by weight is preferred. Paper and fiberglass elements generally have proven unsatisfactory. Filter size in a properly operated glycol system should be 5-10 μm . Larger sizes 25-50 μm may be required during startup and in dirty service.

It may be impossible to judge the effectiveness of filtration by color alone. Even well filtered glycol probably will be black. But, removal of most of the solids will reduce corrosion, plugging and deposits in the reboiler, and may reduce foaming losses. Good filtration is critical for satisfactory performance. It is desirable to measure the pressure differential across the filter and change the element when it reaches about 170 kPa [25 psi].

The use of a carbon purifier downstream from the filter often is recommended. This will produce essentially water-white glycol. Maintenance of this color has proven desirable because it tends to increase dehydration efficiency and minimum foaming, a major source of glycol loss.

Aromatic hydrocarbons are often present in the rich glycol entering the regenerator and will be adsorbed on the carbon filter. They will quickly reach equilibrium loading on the carbon filter although it is likely they are eventually displaced (to some extent) by heavier hydrocarbons. Because of this high aromatic content, changeout of carbon filters requires special precautions to avoid unnecessary exposure of benzene and toluene to workers.

In some units the carbon filter is installed in the lean TEG stream upstream of the contactor to avoid significant aromatic loading.

Coal-based activated carbon should be used because wood-based charcoal tends to break up in use. This carbon can be placed in a metal canister or fill a vessel. In either case, good screens are needed to prevent carbon loss into the system. Said carbon particles, much like iron sulfide, tend to promote a stable foam.

Glycol filters are only effective when used. In especially dirty glycol systems, filters are often bypassed to avoid frequent filter change-out. The problems with this should be obvious. If filter plugging is

excessive, try larger filter size and look for source of problem (e.g., poor inlet separation, degradation, corrosion products, etc.).

PUMPS

A number of types of positive displacement pumps have been used in glycol service. In small units, variations of chemical feed pumps have been used. A Kimray pump of the type shown in Figure 18.10 is used commonly when a gas-powered pump is desired. They have a unique power recovery feature that minimizes the amount of power gas required. In many cases, the exhaust power gas can be used as stripping gas or burned as fuel.

Electric powered plunger type, triplex pumps also are used commonly. These provide a steady flow rate to the absorber. Hardened or chrome plated plungers usually are recommended. Scoring and pitting has been a problem. Piston speeds should be kept below 0.6 m/s [120 ft/min].

INLET SEPARATION

You cannot afford the dehydrator if you cannot afford to place an effective separator on the gas inlet. Salt water will enter the reboiler, evaporate, and coat the walls (causing fire-tube failure) or fill up the unit. This happens frequently on wells that supposedly do not produce salt water "officially."

A good separator also should remove the bulk of the compressor oil, drilling mud, corrosion inhibitors, pipeline dirt, formation solids, and the like, which are somewhat incompatible with glycol unit operation. A full-sized unit meeting the criteria in Chapter 11 should be used.

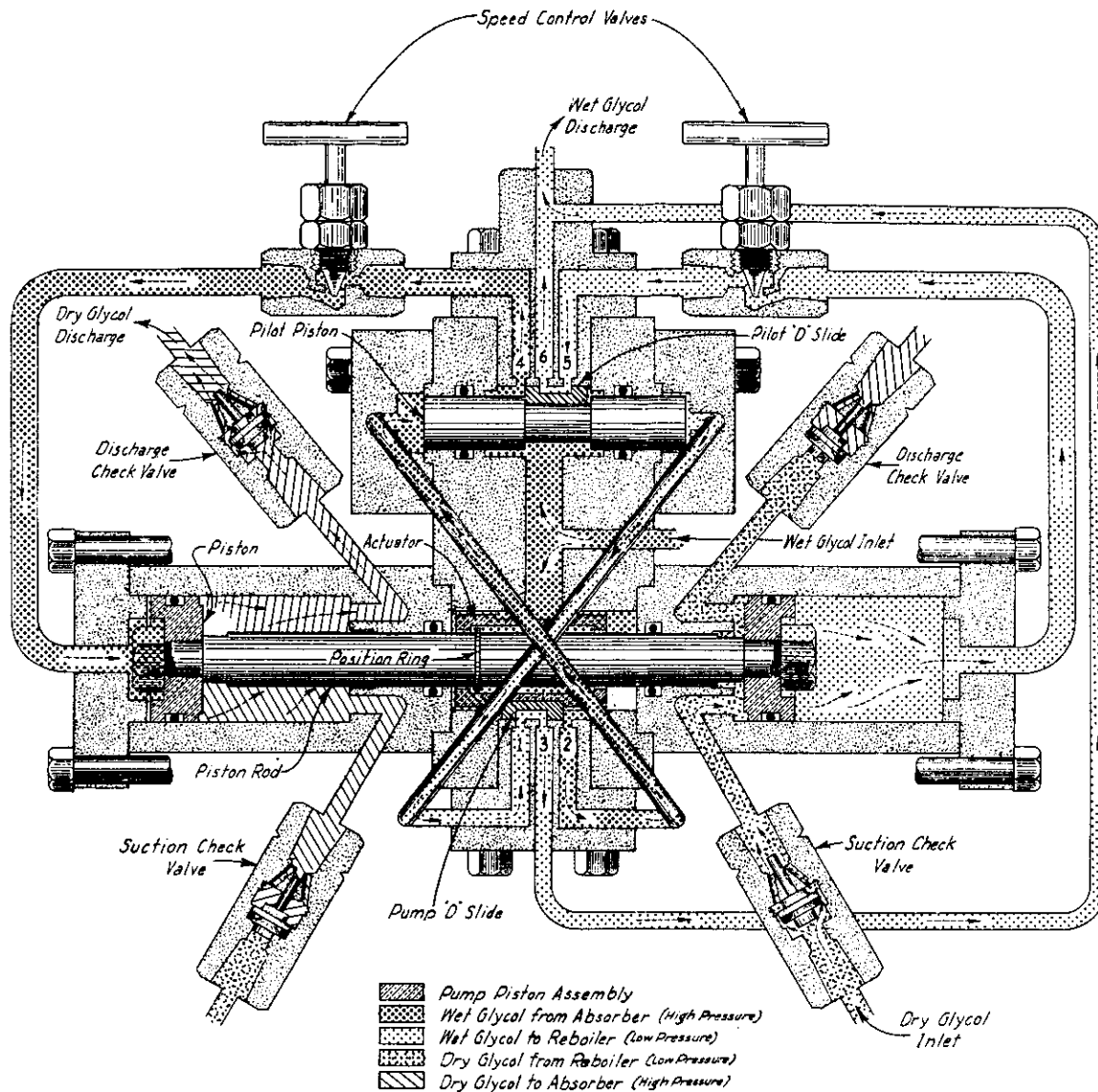
Placing the separator in the bottom of the glycol absorber may be satisfactory but seldom is there enough height to provide performance equivalent to a regular separator. Consider this carefully and do not be too idealistic. In some circumstances this approach represents poor economy.

OPERATING PROBLEMS

The glycol unit should be essentially trouble-free. It seldom is. Many of the problems stem from inadequate design and/or operational faults. The basic simplicity of the unit and the availability of "standard" units tends to obscure the need for attention to mechanical design details. The glycols are very reactive chemically and need to be protected from contamination.

One common symptom of many problems is excess glycol loss. This loss is due to one, or a combination, of the following:

1. Foaming
2. Degradation
3. Salt plugging the regenerator still column
4. Inadequate mist extraction
5. Inadequate absorber design for flow conditions
6. Loss of glycol from pinholes in a gas-glycol coil in the top of the absorber or in a chimney tray above a separator section in the bottom of the absorber
7. Spillage of glycol or pump leakage
8. Lean glycol to absorber is too hot - *Subcooling glycol*



OPERATION:

The Kimray glycol pump is double acting, powered by Wet Glycol and a small quantity of gas at absorber pressure.

Wet Glycol from the absorber flows through port #4 and is throttled through the SPEED CONTROL VALVE to the left end of the Pump Piston Assembly, moving this assembly from left to right. Dry Glycol is being pumped from the left cylinder to the absorber while the right cylinder is being filled with Dry Glycol from the reboiler. At the same time Wet Glycol is discharging from the right end of the Pump Piston Assembly to a low pressure or atmospheric system.

As the Pump Piston Assembly nears the end of its stroke, the POSITION RING on the PISTON ROD contacts the right end of the ACTUATOR. Further movement to the right moves the ACTUATOR and PUMP "D" SLIDE to uncover port #1 and communicate ports #2 and #3. This

exhausts Wet Glycol from the left end of the PILOT PISTON through ports #2 and #3 to the low pressure Wet Glycol system. At the same time port #1 (which was communicated with port #3) admits Wet Glycol to the right end of the PILOT PISTON. This causes the PILOT PISTON and PILOT "D" SLIDE to be driven from right to left.

In its new position the PILOT "D" SLIDE uncovers port #5 and communicates ports #4 and #6. This exhausts Wet Glycol from the left end of the Pump Piston Assembly through ports #4 and #6 to the low pressure Wet Glycol system. Port #5 (which was communicated with port #6) now admits Wet Glycol through the right hand SPEED CONTROL VALVE to the right end of the Pump Piston Assembly.

The Pump Piston Assembly now starts the stroke from right to left. It follows the above procedure with reversed directions of flow.

Figure 18.10 Operation of the Kimray Glycol Pump

Glycol likes to foam. It will foam whenever allowed to. Ordinary foaming may not be critical if the unit is carefully designed. Any foam tends to be more stable when aromatics and/or sulfur compounds are present. Metallic sulfides and sulfites, and degradation products, all contribute to the problem.

Foams are only broken using surface and time, or chemicals. Tray spacing must be large enough so that foam cannot fill up the space between trays and form a continuous liquid phase. A mist extractor does not break foam effectively. Once foam fills the absorber, there is a continuous liquid phase for glycol to go out overhead.

Use of an antifoam agent can reduce the problem. Figure 18.11 shows the effect of adding trioctyl-phosphate to maintain a concentration of 500 ppm. There are many antifoam agents available. One that works in one unit may not work equally well in another. Some trial-and-error testing of an antifoam agent, and concentration of that agent, is often necessary.

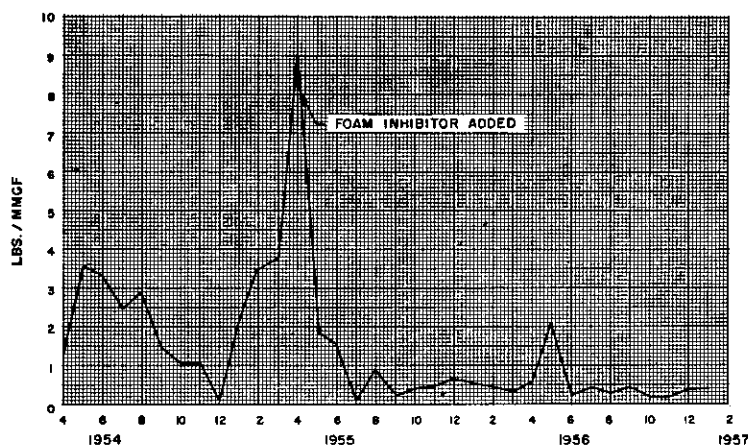


Figure 18.11 Possible Effect of Foam Inhibitors in TEG Systems^(18.8)

Avoid adding too much antifoam agent. If too much is added it may accelerate foaming. Set up a careful control policy so operators keep unit concentration within the limits specified.

Degradation is a natural occurrence and is accelerated in the presence of sulfur compounds. The answer is effective filtration. Degradation products contribute to foaming but they also are major sources of corrosion problems. The best filtration system uses a regular filter to remove the large "chunks" and then an activated carbon filter to remove hydrocarbons as well as fine contaminants that pass through the first filter. The initial cost is higher but the carbon filter may offer a favorable benefit cost/ratio.

Salt is a continuing problem. Good separation ahead of the absorber is mandatory. Any salt arriving at the regenerator deposits either in the still column or in the reboiler. It is common for packed still columns to plug up to the point glycol is lost overhead. If this does not occur, salt can plug the reboiler and cause failure. Not providing good separation is inexcusable.

The water vapor in gas is relatively fresh but is slightly saline. NaCl is soluble in TEG to some degree. At 50°C about 3.3 kg will dissolve in 100 kg of TEG. So, some salt is always present. The soluble salt hydrolyzes to HCl and lowers the pH of the glycol.

Corrosion-Erosion

Glycols are very reactive with sulfur compounds. The resultant materials tend to polymerize and form "gunk" which is very corrosive. Also, the glycol pH becomes lower. Corrosion inhibitors alone cannot solve

the problem satisfactorily. The real solution is good mechanical design and good filtration supplemented by a corrosion inhibitor.

Good design involves factors like control of fluid velocities, long radius ells and a host of little details that too seldom are done properly. In many cases, good mechanical design will eliminate the need for expensive alloy steels.

If feasible to do so, the glycol pH should be maintained above 6.0. Some become so preoccupied keeping it at 7.0 or above that they add copious amounts of caustic, sodium carbonate and the like to the unit. The result is seldom satisfactory. Adjustment of pH is proper but the cure can be worse than the disease if it is overdone.

Corrosion inhibitors which plate out on metal surfaces and form a film can be effective in minimizing corrosion. Many materials are available. A product called Nacap has been used often. Some have used the amines effectively.

Figure 18.12 shows one result of using a corrosion inhibitor in a glycol system.

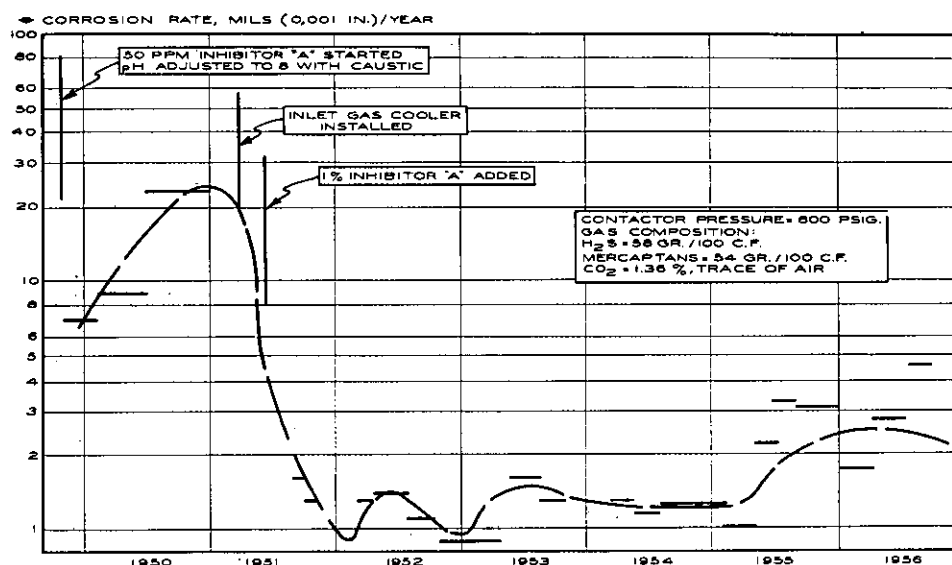


Figure 18.12 Possible Effect of Corrosion Inhibitors in TEG Systems^(18.8)

Notice that corrosion was not eliminated; it merely was reduced to a satisfactory level. In a corrosive environment, the total elimination of corrosion is an unrealistic goal. The proper goal is reducing it to economically tolerable levels.

Figure 18.13 shows the solubility of H_2S in TEG.^(18.9) This is true absorption that takes place in the absorber. It lowers pH and provides a mechanism for reactions. Figure 18.14 shows solubility of CO_2 in a 96.5 wt% TEG solution.^(18.10) Solubility of CO_2 in pure TEG is approximately 20% higher. Reference 18.11 provides additional data on the solubilities of H_2S and CO_2 , as well as C_1 , C_2 and C_3 in TEG.

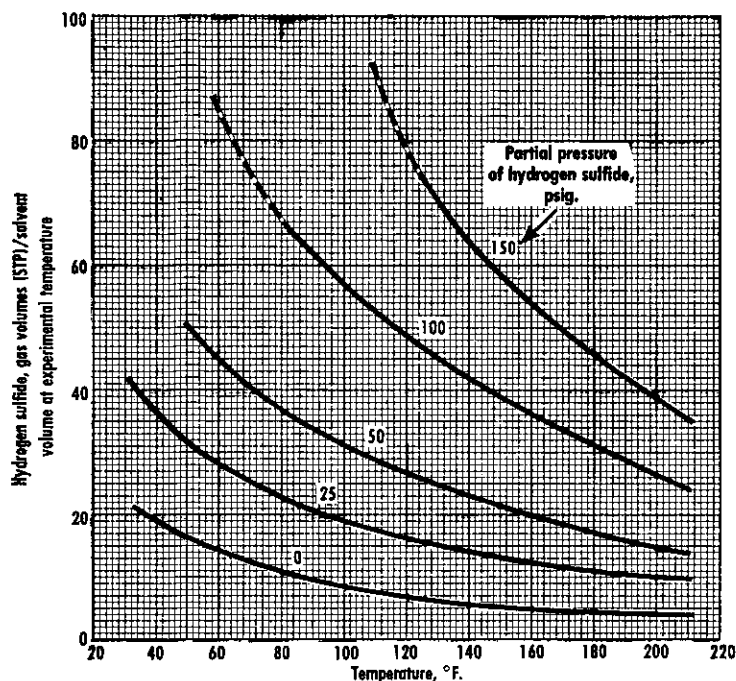


Figure 18.13 Solubility of Hydrogen Sulfide in TEG at Various Partial Pressures [30-230°F]^(18.8)

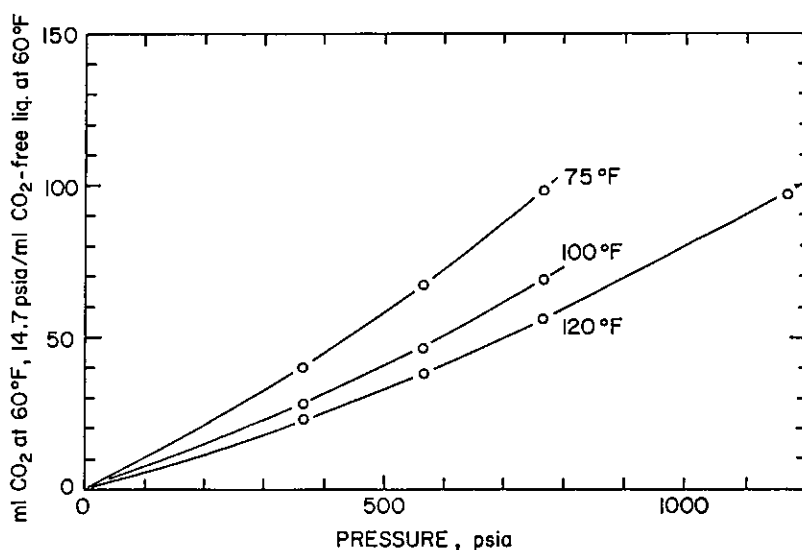


Figure 18.14 CO₂ Solubility in TEG 3.5 wt% H₂O

AROMATIC ABSORPTION

The affinity for aromatics by TEG has long been recognized. The UDEX process was used for many years in refineries and chemical plants to extract aromatic hydrocarbons from paraffins with TEG.

In gas dehydration service, TEG will absorb limited quantities of aromatic hydrocarbons (benzene, toluene, ethylbenzene and xylene) from the gas. these components are often abbreviated as BTEX. Quantifying the absorption levels has become more important in recent years due to increased restrictions on aromatic emissions from glycol units. To date, little published equilibrium data is available.^(18.14,18.15)

Based on data from Reference 18.15, predicted absorption levels for BTEX components vary from 5-10% for benzene to 20-30% for ethylbenzene and xylene. Absorption is favored at higher pressures and lower temperatures, and increased TEG concentration and circulation rate.

The bulk of absorbed aromatics will be vented with the water vapor at the top of the regenerator. In some cases these aromatics can be condensed and recovered, however the effectiveness of condensation declines rapidly with increasing stripping gas rates.

In these cases, many operators use a partial condenser to condense the water and pipe the volatile hydrocarbons to an incinerator or to the reboiler as fuel. Back pressure on the regenerator is minimized by the use of an eductor – which is often the fuel gas valve on the reboiler.

Another mitigation measure which looks promising is to strip the aromatics from the TEG with a small stripping gas stream at the flash tank. The stripping vapors are then at a sufficiently high pressure to allow recycling to a compressor or use in a remote fuel gas system.

Others have proposed using membranes or activated carbon but these methods are not yet commercial.

This problem is one which requires careful attention in the design phase. Environmental considerations are increasingly driving the selection and operation of process equipment. In some cases the use of a dry desiccant unit (albeit at a higher capital cost) may be a lower cost alternative to glycol given the environmental impact of these BTEX emissions.

CONTROLS

The control system usually is rather simple. A regular displacement type level control is used on the absorber and flash tank. A back-pressure valve on the gas outlet may be required to maintain a stable pressure in the absorber. A manual bypass around the absorber is desirable to aid in the startup and shutdown of the unit. In some units standby methanol injection is provided so that gas flow can be maintained when the unit is down.

In a gas-fired reboiler the use of dual thermostats is very satisfactory. With other heat sources, a controller using only proportional mode usually is satisfactory. Figure 18.15 shows two systems used for waste heat control.

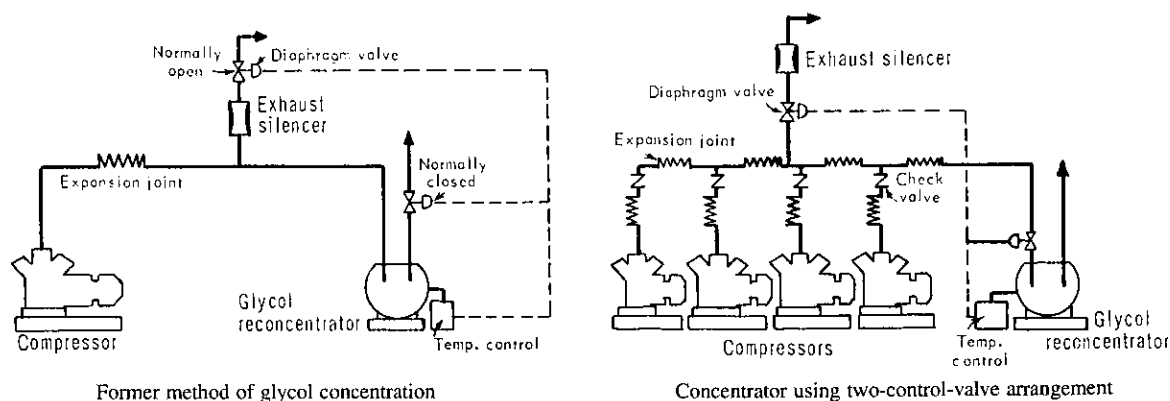


Figure 18.15 Two Systems Using Waste Heat for Regeneration^(18.12)

The right portion of Figure 18.15 also shows some of the controls employed.

Figure 18.16 is a schematic of a glycol unit on a platform involving the separation of water condensate from gas, free water disposal and gas dehydration. A remote monitoring and control system is used for routine operations.

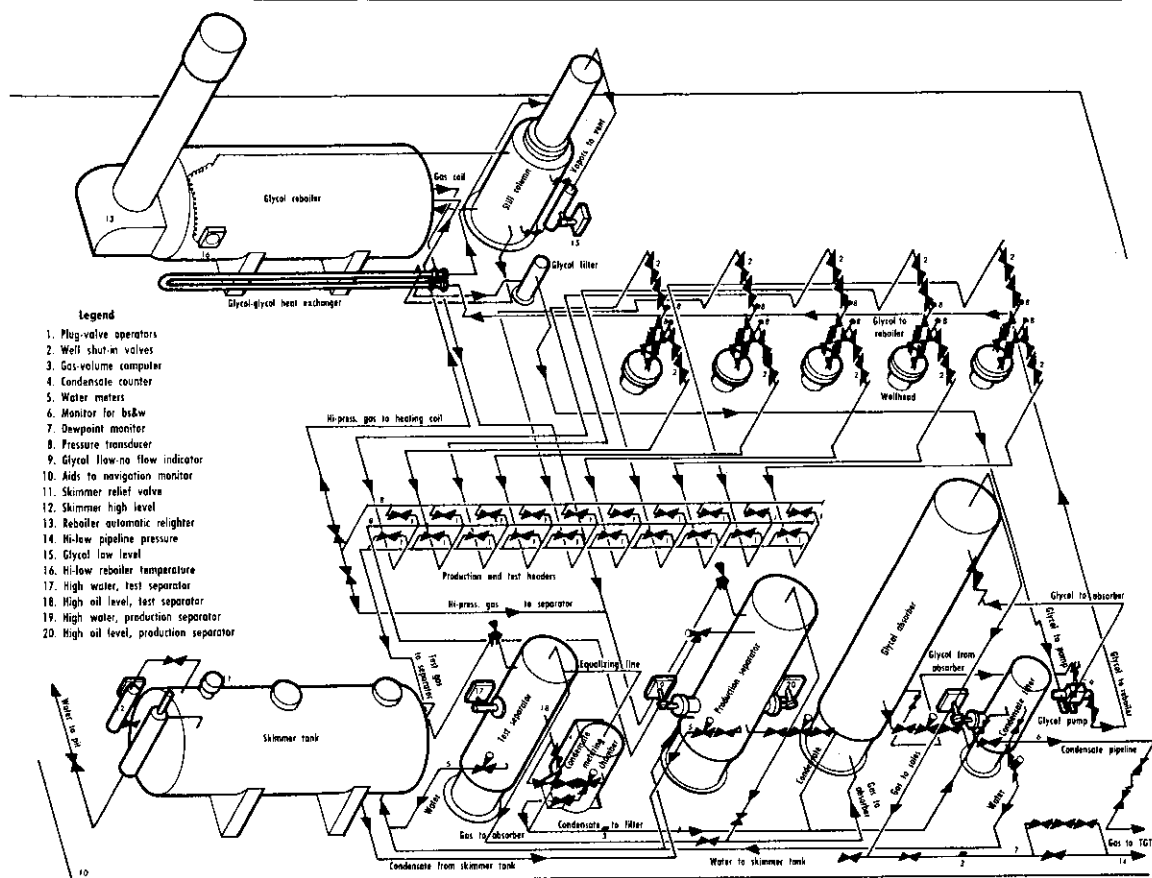


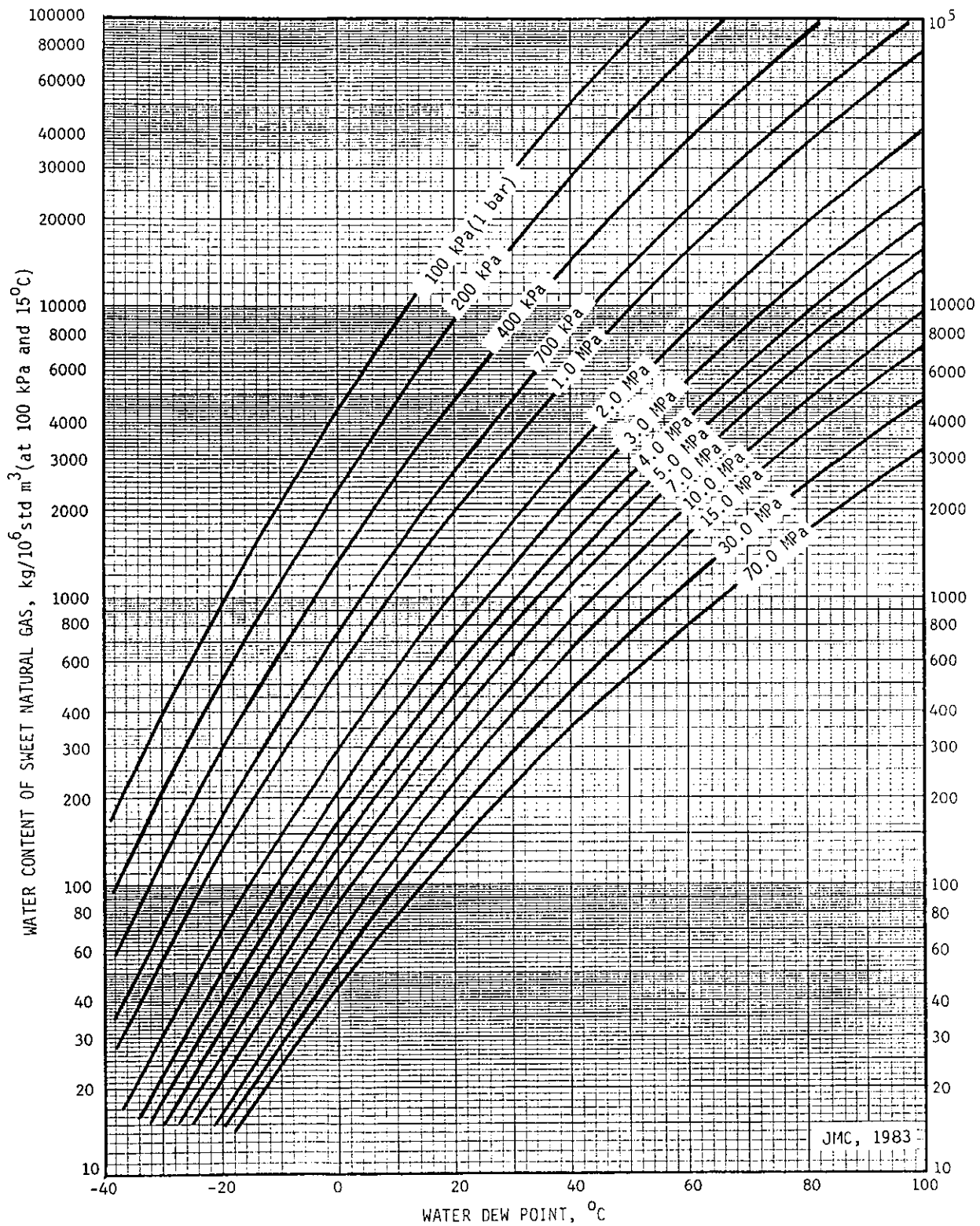
Figure 18.16 Layout of One Offshore Platform for Dehydration^(18.13)

REFERENCES

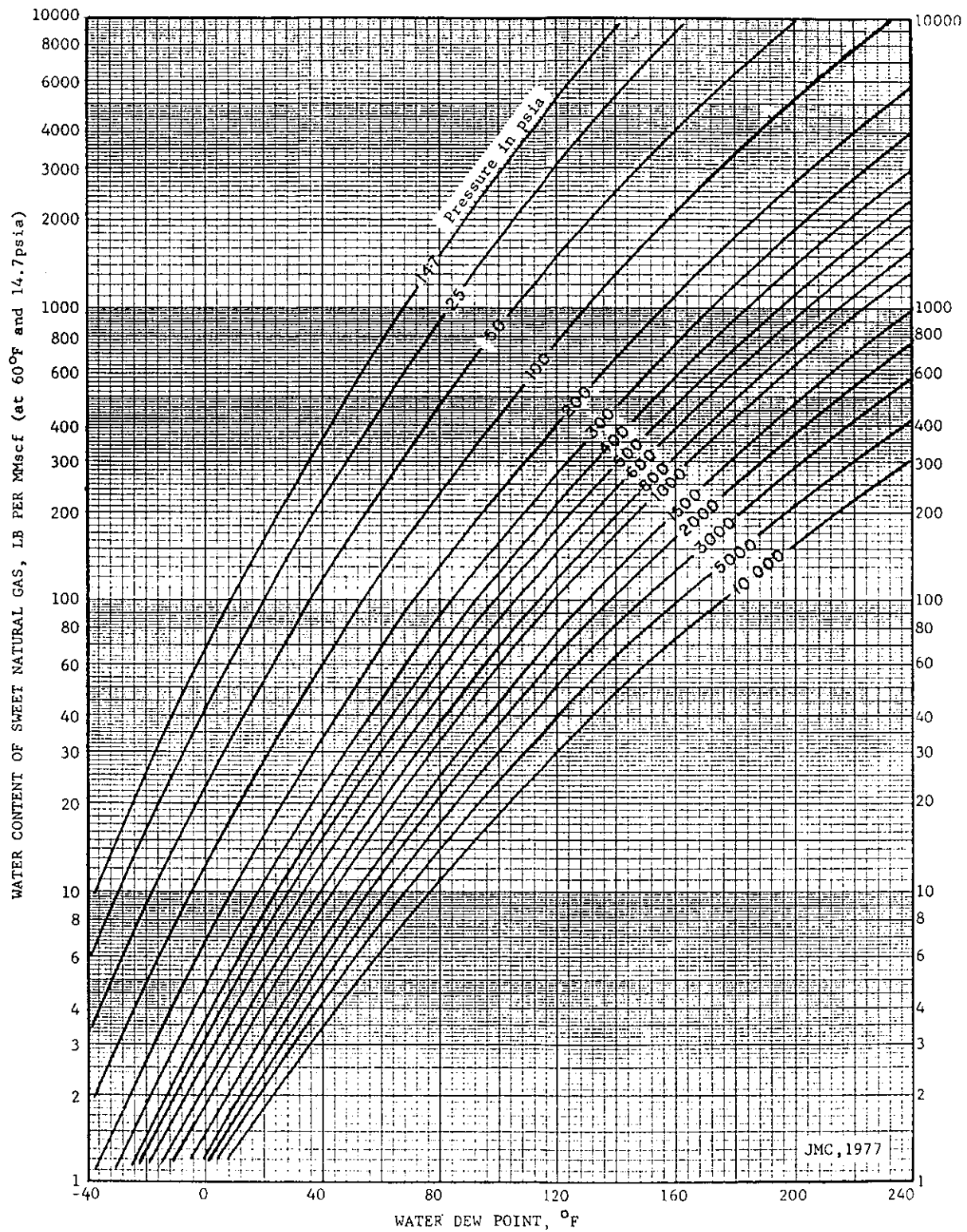
- 18.1 Worley, M. S., Gas Cond. Conf., Univ. of Oklahoma (April 1967).
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APPENDIX 18A

GLYCOL PROPERTIES AND WATER CONTENT CHARTS FROM CHAPTER 6.



Water Content of Sweet, Lean Natural Gas. (Metric)



Water Content of Sweet, Lean Natural Gas. (English)

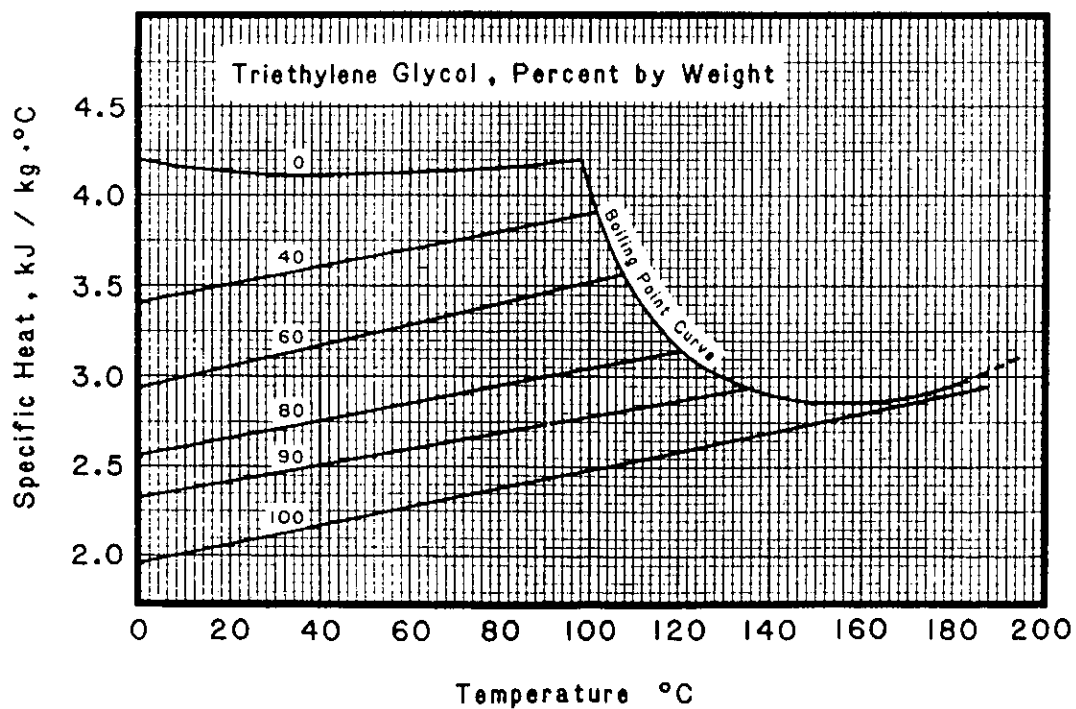
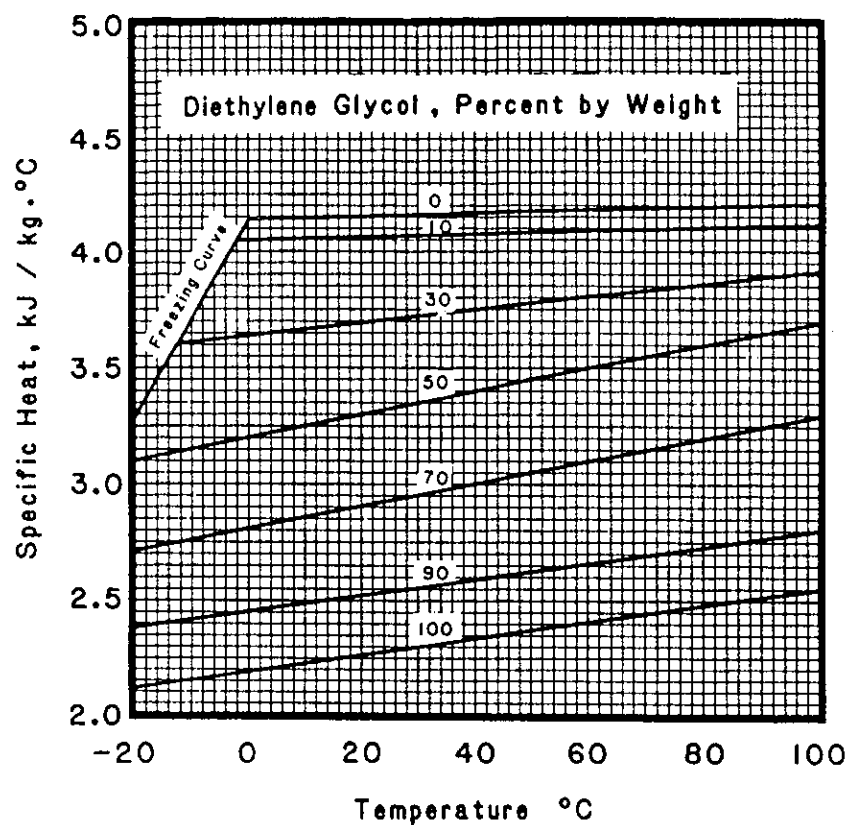
Physical Properties of Glycols

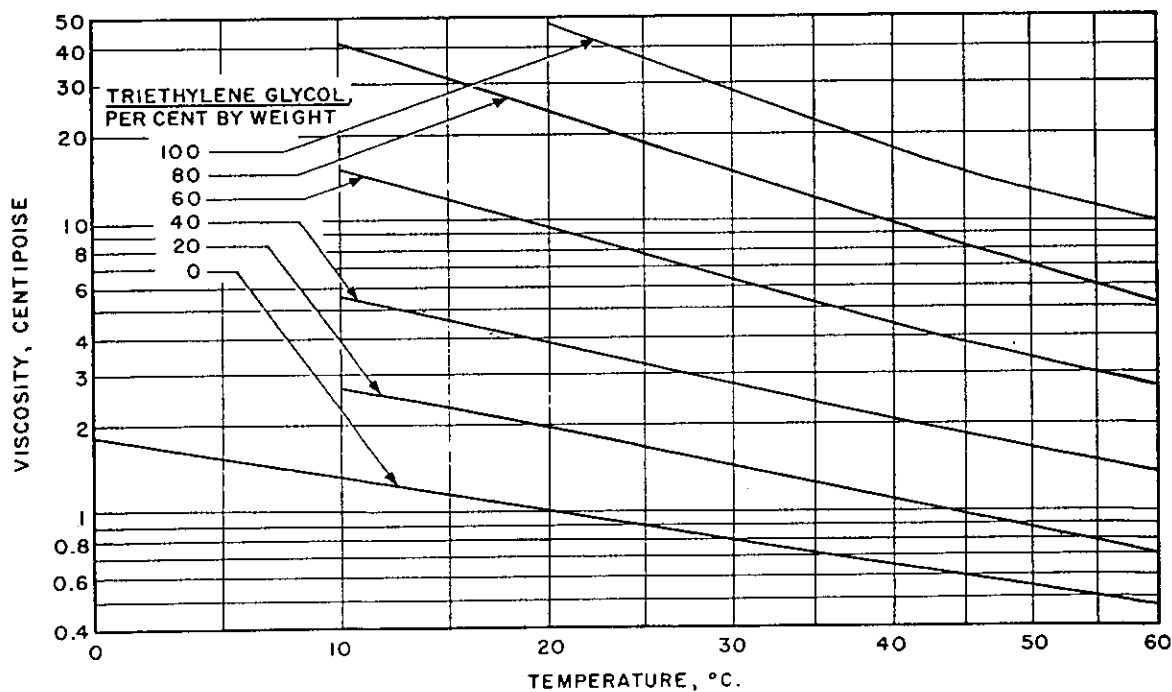
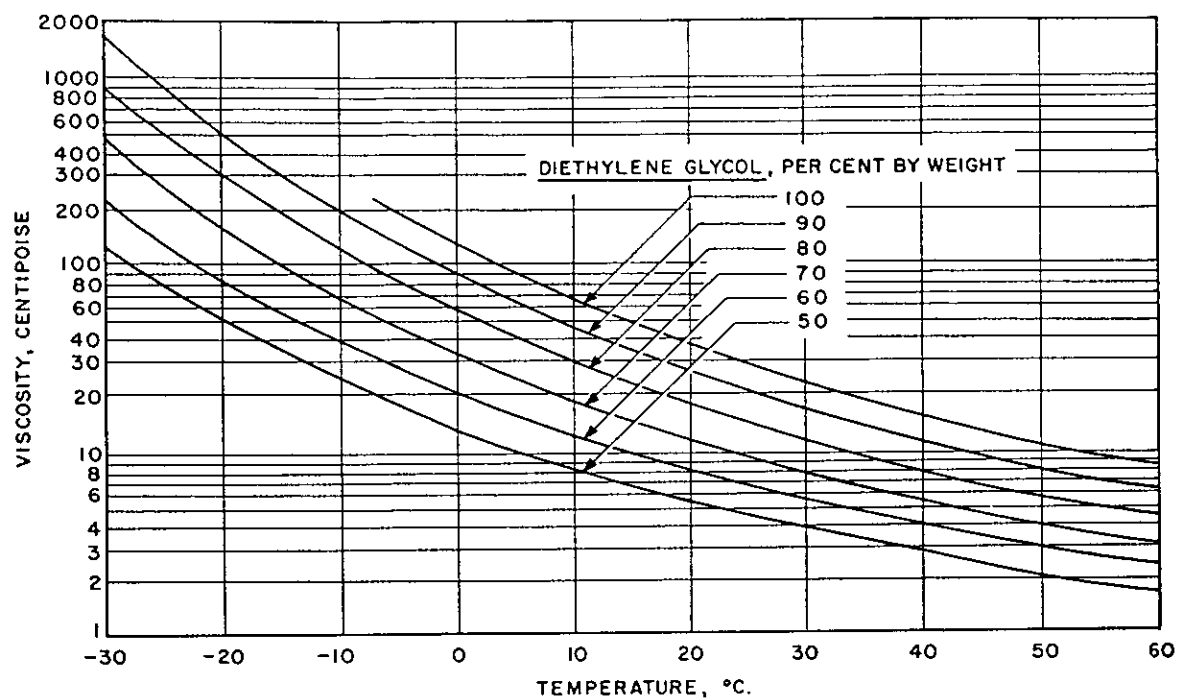
	EG	DEG	TEG
Formula	$C_2H_6O_2$	$C_4H_{10}O_3$	$C_6H_{14}O_4$
Molecular Weight	62.1	106.1	150.2
Boiling Point at 760 mm Hg °F	387.1	472.6	545.9
Boiling Point at 760 mm Hg °C	197.3	244.8	288
Vapor Pressure at 25°C, mm Hg	0.12	0.01	0.01
Density at 25°C, g/cm ³	1.110	1.113	1.119
at 60°C, g/cm ³	1.085	1.088	1.092
Pounds per Gallon at 25°C	9.26	9.29	9.34
Freezing Point, °C	-13	-8	-7
Pour Point, °C	-	-54	-58
Viscosity in Centipoises at 25°C	16.5	28.2	37.3
At 60°C	4.68	6.99	8.77
Surface Tension at 25°C Dynes/cm	47	44	45
Refractive Index at 25°C	1.430	1.446	1.454
Specific Heat at 25°C	0.56	0.55	0.53
Flash Point, °C (COC)	116	138	160
Fire Point, °C (COC)	119	143	166

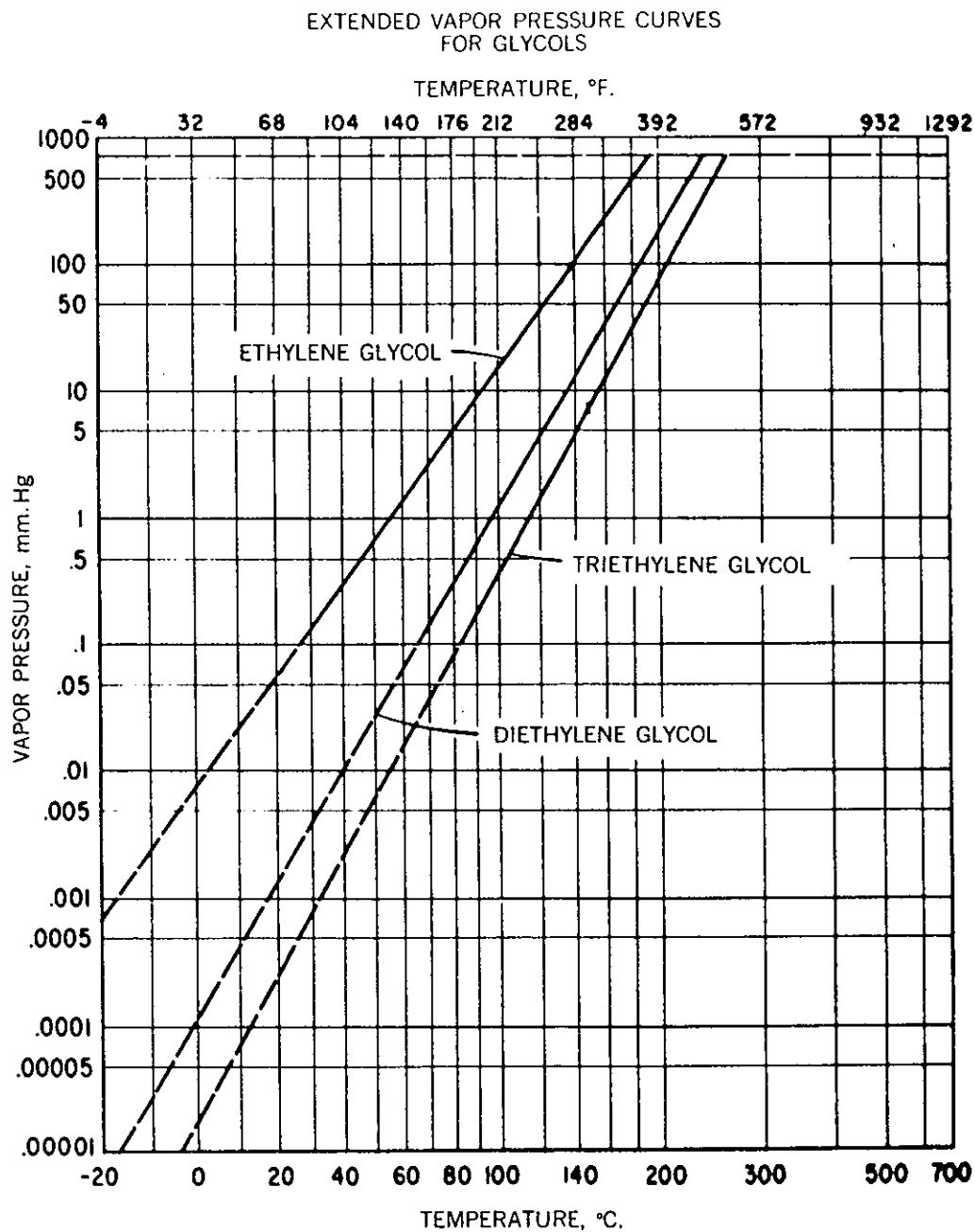
HEAT TRANSFER FACTORS FOR GLYCOL

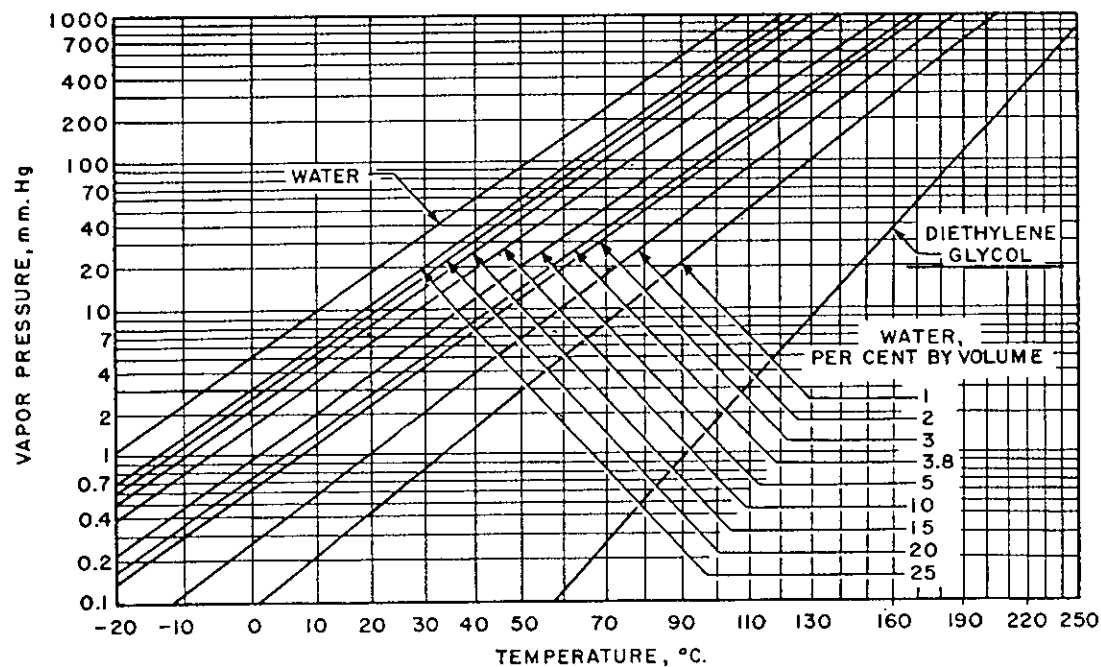
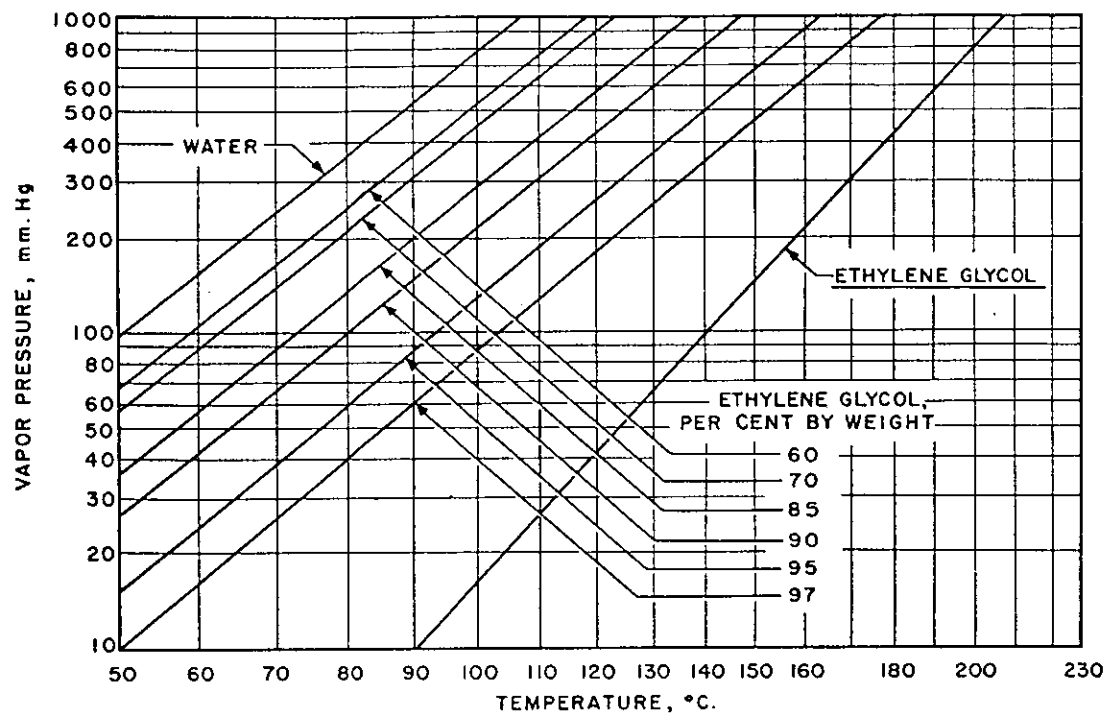
Material	Liquid Heating or Cooling		Vapor Heating or Cooling	Condensing Vapor	Boiling Liquids
	Re < 2100	Re > 2100			
Ethylene Glycol	0.49	0.12	0.60	0.18	0.30
Diethylene Glycol	0.50	0.11	0.60	0.17	0.30
Triethylene Glycol	0.49	0.09	0.60	0.15	0.30

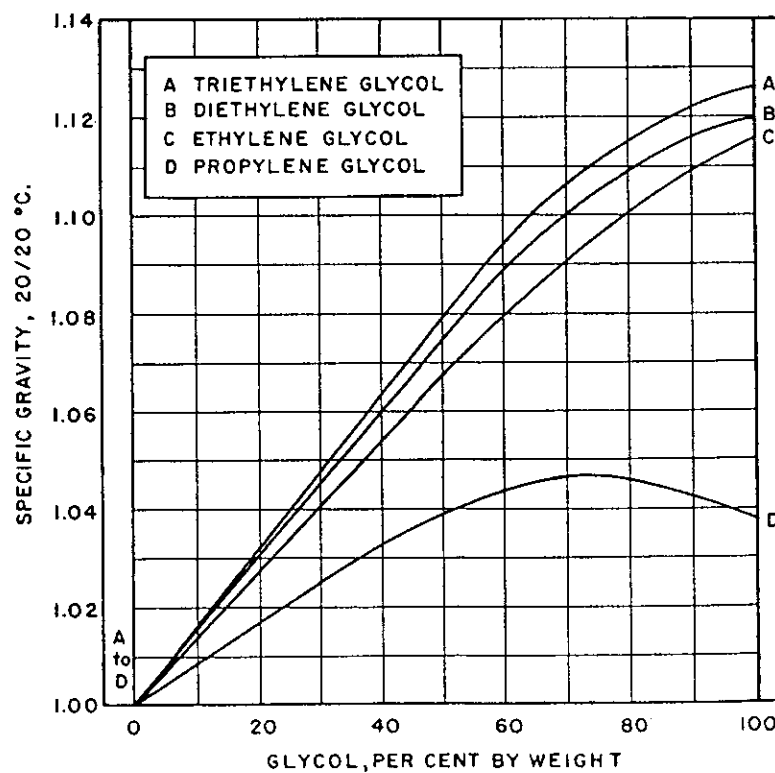
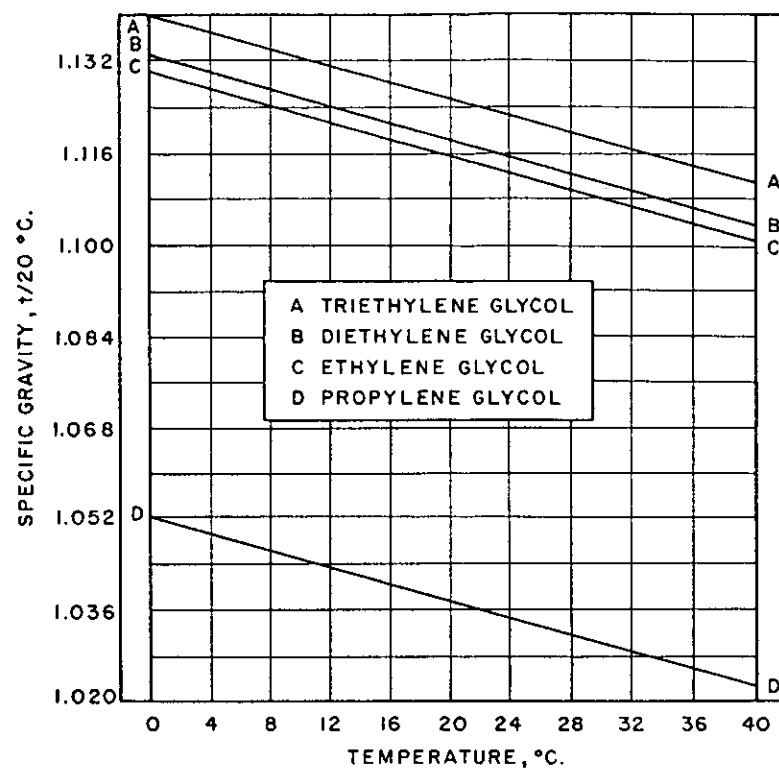
The water factor = 1.0. Multiply heat transfer coefficients for water by the factor shown to obtain glycol coefficient for the service involved.

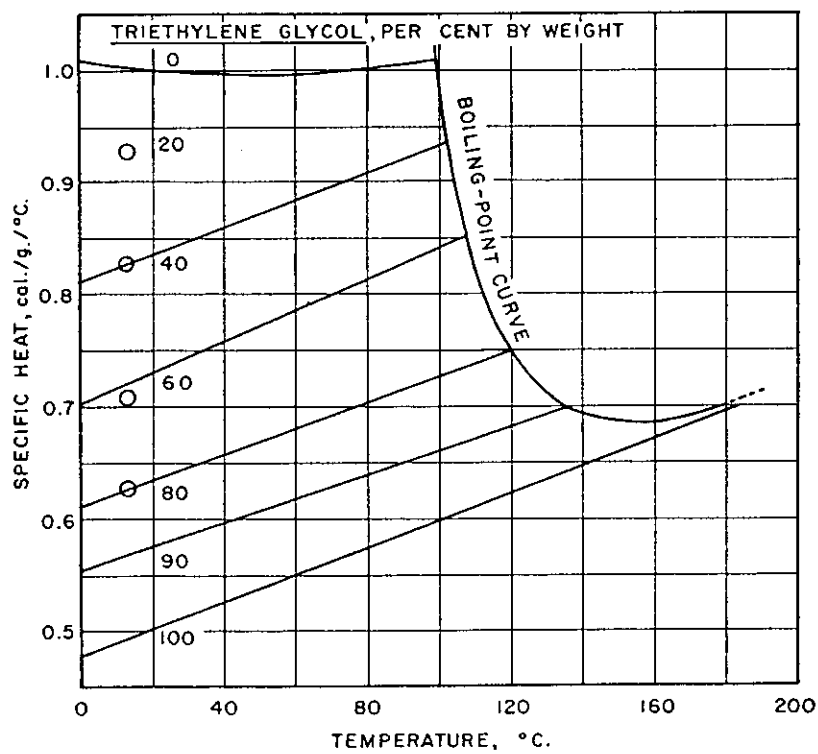
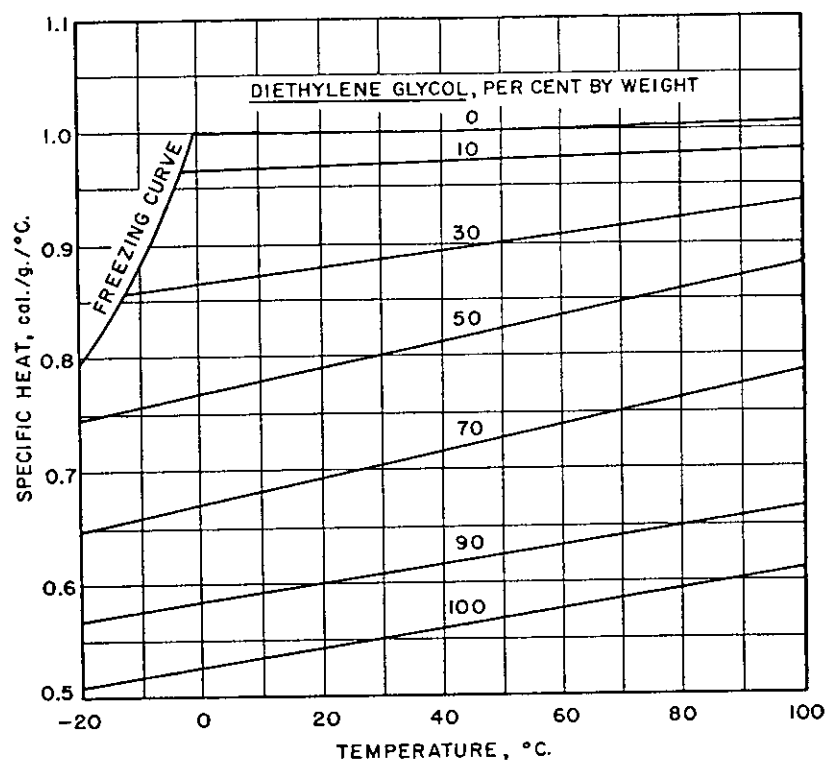


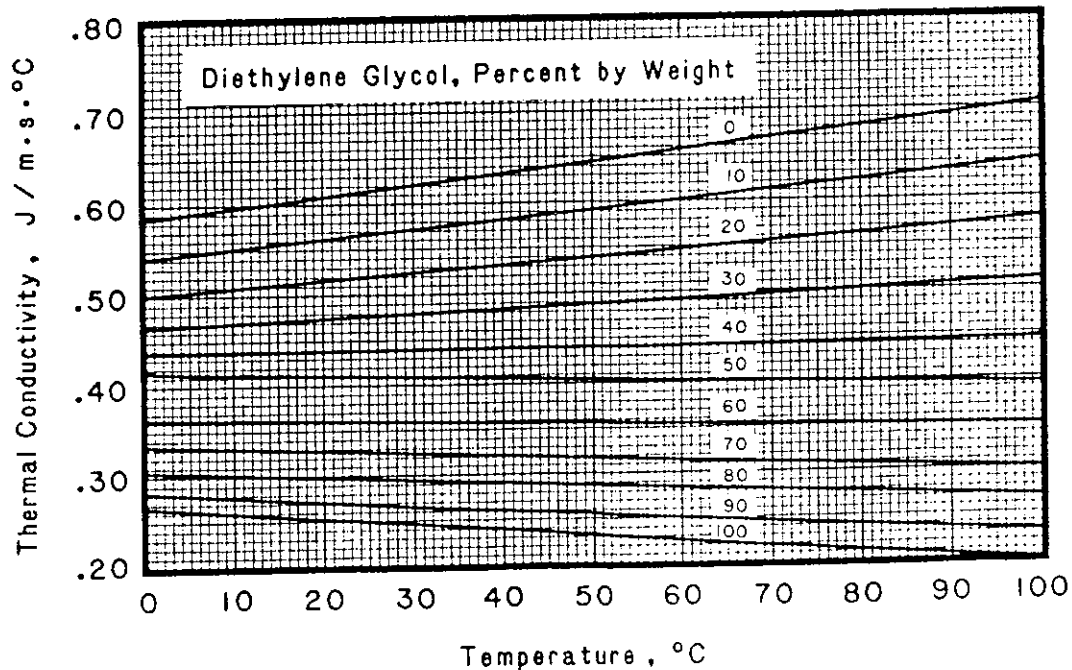
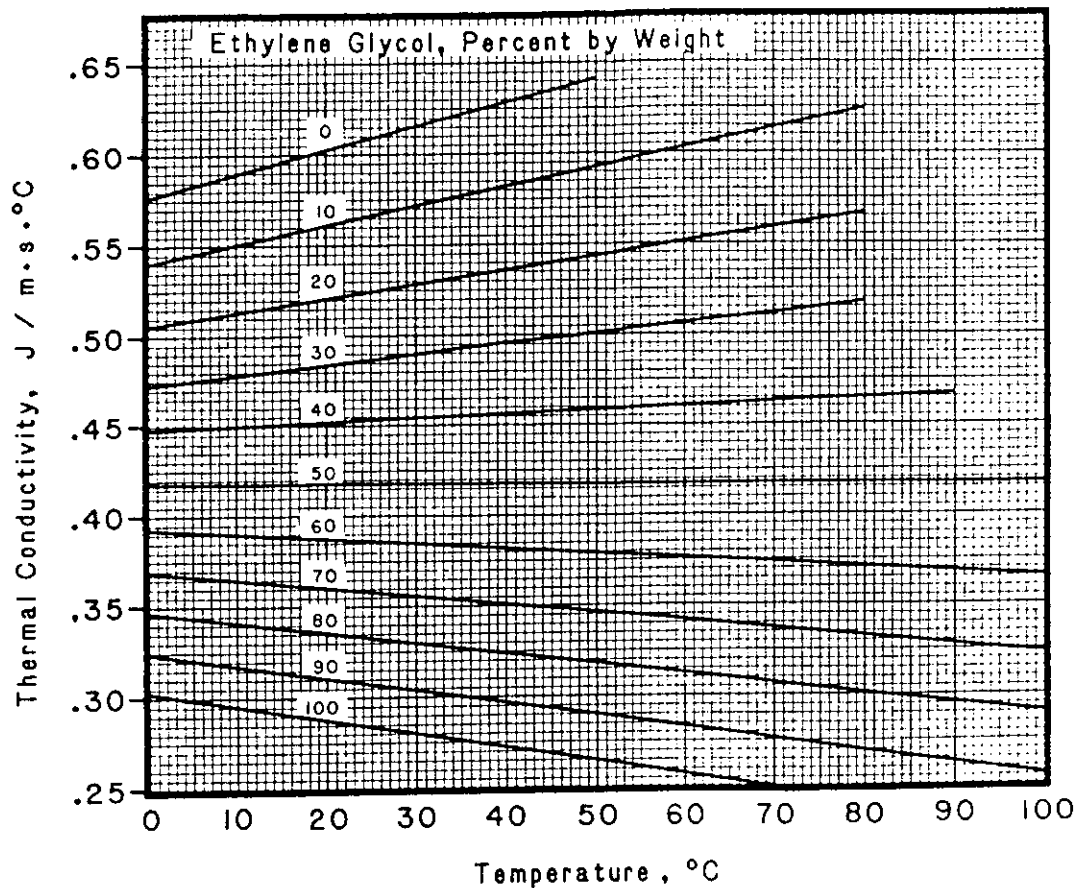


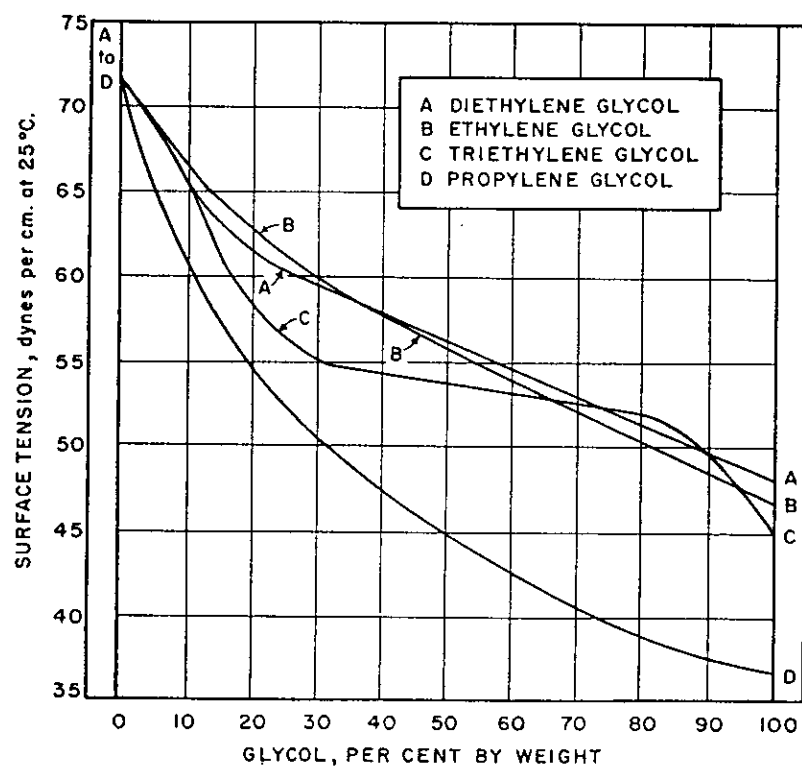
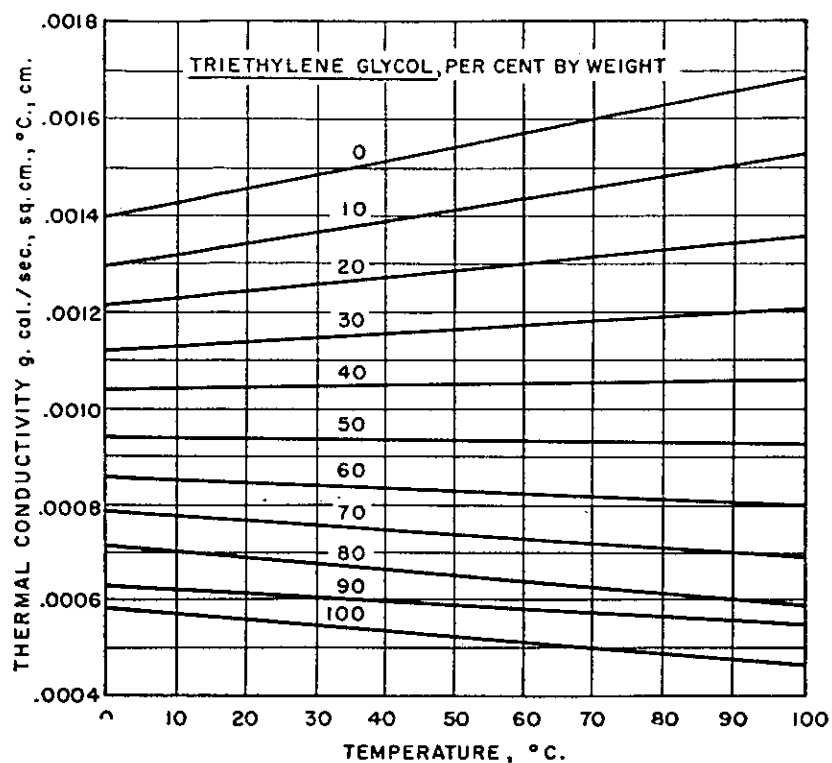












NOTES:

19

ADSORPTION DEHYDRATION AND SWEETENING

Adsorption describes any process wherein molecules from the gas are held on the surface of a solid by surface forces. Adsorbents may be divided into two classes – those which owe their "activity" to surface adsorption and capillary condensation, and those which react chemically. The latter group finds limited application in natural gas processing and will not be discussed herein. The former (physical adsorption) requires use of an adsorbent material which probably has the following characteristics:

1. Large surface area for high capacity.
2. Possesses "activity" for the components to be removed.
3. Mass transfer rate is high.
4. Easily and economically regenerated.
5. Good activity retention with time.
6. Small resistance to gas flow.
7. High mechanical strength to resist crushing and dust formation.
8. Fairly cheap, non-corrosive, non-toxic, chemically inert and possesses a high bulk density.
9. No appreciable change in volume during adsorption and desorption, and should retain strength when "wet."

Any commercial adsorbent will have a total surface area of 500 to 800 sq. meters per gram [2 400 000 to 3 900 000 sq. ft. per lb.]. One pound can easily be held in your cupped hands. This fantastic appearing area is only achieved by producing a material with large interior surface resulting from capillaries or a crystalline-type lattice. The exterior surface of the particles is almost negligible.

The materials which meet the above requirements may be divided into several general categories:

Bauxite – naturally occurring mineral composed primarily of Al_2O_3 .

Alumina – a purer, manufactured version of bauxite.

Gels – composed largely of SiO_2 or alumina gel; manufactured by chemical reaction.

Molecular Sieves – a calcium-sodium aluminosilicate (zeolite).

Carbon (charcoal) – a carbon product treated and activated to have adsorptive capacity.

(Only listed are those materials commonly used for bulk treating.)

All but carbon are used for dehydration. Carbon has desirable properties for hydrocarbon removal and adsorption of certain impurities but possesses negligible water capacity. The first four desiccants are listed in the order of their capital cost. As might be expected, the higher-priced materials possess desirable

characteristics that may justify their cost. For one thing, capacity increases with cost, although it is not proportional to it. The final choice must be based on equipment costs, service life, applicability to process needs, etc. Undue preoccupation with initial cost can therefore be misleading. On the other hand, purchasing molecular sieves, the most versatile of the adsorbents, might be equally illogical because of their higher cost. Choosing the proper adsorbent for a given service inevitably involves a rational compromise between cost and need. Too often this choice is made poorly (bias and habit seemingly predominating). The purchaser should actively participate in this choice. (Vendors also suffer from bias and habit.) For routine applications where several materials will suffice, the choice is often not automatic. In most adsorption plants proper design and operation is more critical than choice of adsorbent. A rather abortive attempt to adapt "standard" units to non-standard service accounts for too many of the problems encountered.

Desiccant Properties

Summary of Typical Desiccant Properties						
Property	Grade 03 Silica Gel	Mobilbead R	Mobilbead H	H-151 Alumina Gel	F-1 Alumina	4A-5A Molecular Sieves
Surface Area, m ² /g	750-830	550-650	740-770	350	210	650-800
Pore Volume, cm ³ /g	0.40-0.45	0.31-0.34	0.50-0.54	0.35	0.21	0.27
Pore Diameter, °A	21-23	21-23	27-28	43	26	(1)
Bulk Density, kg/m ³	721	785	721	833-881	801-881	689-721
Sp. Ht., kJ/(kg·°C)	0.92	1.05	1.05	0.84	1.0	0.96

Notes: (1) Types 4A and 5A contain cavities 11.4°A in diameter with circular openings 4.2°A in diameter (opening size for adsorption). 10⁸ angstroms (°A) = 1 cm.

(2) The values in Column 1 of the above table can be converted to English units using the following conversion factors:

Surface area, ft²/lbm = 4885 (m²/g)

Pore Volume, in³/lbm = 27.7 (cm³/g)

Bulk Density, lbm/ft³ = 0.0624 (kg/m³)

Specific Heat, Btu/(lbm·°F) = 0.24 (kJ/kg·°C)

The potential capacity per unit volume is a product of bulk density times the available area for adsorption. In essence, monolayer adsorption occurs. An examination of the above shows why the gels have a higher effective capacity than the aluminas.

The pore opening at the surface of the desiccant must be large enough to admit the molecules being adsorbed to the interior of the particle where most of the surface area exists. In the internal pores of the gels are capillaries of the diameter range shown. With molecular sieves, the internal pores are crystalline cavities larger than the openings on the surface.

In the table following are shown the *nominal diameter* of common molecules involved in hydrocarbon system adsorption. This is called the nominal diameter because the molecules are not spheres and their ability to enter a given size opening depends on their direction of approach. Also, they are flexible and can "squeeze" through an opening to some degree.

Molecule	Nominal Diameter, °A	Molecule	Nominal Diameter, °A	Molecule	Nominal Diameter, °A
Hydrogen	2.4	Hydrogen Sulfide	3.6	Propane	4.9
Carbon Dioxide	2.8	Methanol	4.4	nC ₄ -nC ₂₂	4.9
Nitrogen	3.0	Methane	4.0	iC ₄ -iC ₂₂	5.6
Water	3.2	Ethane	4.4	Benzene	6.7

The various commercial desiccants can be divided into three broad categories: alumina, gel and molecular sieves. Within each are a series of trade names.

Alumina is a hydrated form of aluminum oxide (Al_2O_3). When manufactured it is essentially iron free. In its natural state (bauxite) it contains varying amounts of iron. It is *activated* by driving off part of the hydrated water adsorbed on the surface. $\text{Al}_2\text{O}_3 \cdot 3\text{H}_2\text{O}$ the hydrated version would be heated to from say $\text{Al}_2\text{O}_3 \cdot 1\text{H}_2\text{O}$, leaving the particle short of its equilibrium water content.

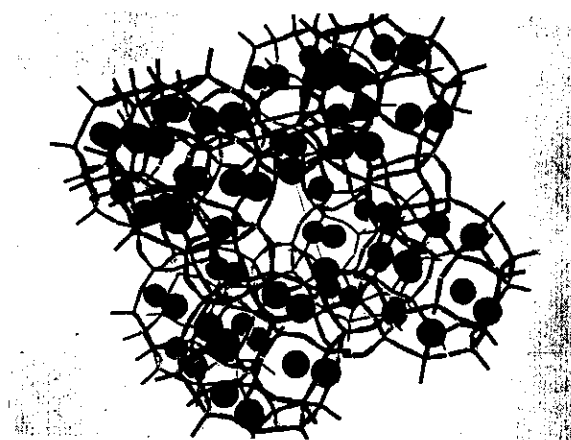
Gel is a granular, amorphous solid. Silica Gel is the generic name for a gel manufactured from sulfuric acid and sodium silicate. It is essentially 100% silicon dioxide (SiO_2). Other gels like *alumina gel* may be largely a form of Al_2O_3 . Other gel type desiccants are some combination of these two.

Molecular Sieves are alkali metal crystalline aluminosilicates very similar to natural clays. 4A molecular sieves are composed of Na_2O , Al_2O_3 and SiO_2 . Types 3A and 5A are produced by ion exchange of about 75% of the Na ions by potassium and calcium ions. Type 10X is produced from 13X by ion exchange of about 75% of the Na ions by Ca ions. All types have a pH of about 10 and are stable in the pH range of 5-12.

The affinity for water is based on the previous environment. However, polarity of the water molecule also plays an important part. Molecular sieves have electric charges on the inner surfaces of the crystal cavities, which are attracted to similar charges on polar molecules. Such molecules, including hydrogen sulfide, ammonia, carbon monoxide, methylamine, and the alcohols, are adsorbed in preference to non-polar molecules. Similarly, molecular sieves show a preference for "unsaturated" hydrocarbons, in which some of the carbon atoms are joined together by double or triple chemical bonds. This is because these compounds contain loosely bound electrons which give them polar characteristics resembling those of water molecules. As an example, if a mixed stream of ethane (a saturated hydrocarbon) and ethylene (an unsaturated hydrocarbon) is passed through a molecular sieve bed, eighty percent of the molecules adsorbed will be ethylene.

The A type sieves have a crystalline zeolite structure consisting of intracrystalline voids as shown at right. All adsorption takes place in these voids. The voids are 11.4 Å in diameter and are connected by openings 4.2 Å in theoretical diameter (pore diameter). The effective pore diameter is determined by the cation and its position in the structure. The maximum diameter of molecules that can enter the crystalline structure and be adsorbed are as follows:

Type	Molecule Diameter - Å
3A - potassium zeolite	3
4A - sodium zeolite	4
5A - calcium zeolite	5
10X - calcium zeolite	8
13X - sodium zeolite	10



The X type sieves vary from the A type in the internal character of the crystalline structure. Their adsorption characteristics are the same. The X type can adsorb all molecules adsorbed by the A type with somewhat higher capacity. 13X can adsorb large molecules such as aromatics.

The selective capacity of molecular sieves for different sizes of molecules is important. To a degree, one can exclude those sizes too large to enter the crystal. This is why a 3A or 4A sieve might be used for drying. Sieves are likewise used for high temperatures because their capacity does not decrease as much as gel or alumina above 38°C. Table 19.1 summarizes the characteristics of regular molecular sieves.

TABLE 19.1
Basic Characteristics of Molecular Sieves

Basic Type	Nominal Pore Diameter (Angstroms)	Available Form	Equilibrium H ₂ O Capacity (% wt)*	Molecules Adsorbed**	Molecules Excluded	Applications
3A	3	Powder 1/16-in. Pellets 1/8-in. Pellets	23 20 20	Molecules with an effective diameter <3 angstroms, including H ₂ O and NH ₃	Molecules with an effective diameter >3 angstroms, e.g. ethane	The preferred Molecular Sieve adsorbent for the commercial dehydration of unsaturated hydrocarbon streams such as cracked gas, propylene, butadiene, and acetylene. It is also used for drying polar liquids such as methanol and ethanol.
4A	4	Powder 1/16 in. Pellets 1/8-in. Pellets 8 x 12 Beads 4 x 8 Beads 14 x 30 Mesh	28.5 22 22 22 22	Molecules with an effective diameter <4 angstroms, including ethanol, H ₂ S, CO ₂ , SO ₂ , C ₂ H ₄ , C ₂ H ₆ , and C ₃ H ₆	Molecules with an effective diameter >4 angstroms, e.g. propane	The preferred Molecular Sieve adsorbent for static dehydration in a closed gas or liquid system. It is used as a static desiccant in household refrigeration systems; in packaging of drugs, electronic components and perishable chemicals; and as a water scavenger in paint and plastic systems. Also used commercially in drying saturated hydrocarbon streams.
5A	5	Powder 1/16-in. Pellets 1/8-in. Pellets	28 21.5 21.5	Molecules with an effective diameter <5 angstroms, including n-C ₄ H ₉ OH**, n-C ₄ H ₁₀ **, C ₃ H ₈ to C ₂₂ H ₄₆ , R-12	Molecules with an effective diameter >5 angstroms, e.g. iso compounds and all 4 carbon rings	Separates normal paraffins from branched-chain and cyclic hydrocarbons through a selective adsorption process.
10X	8	Powder 1/16-in. Pellets 1/8-in. Pellets	36 28 28	Iso paraffins and Olefins, C ₆ H ₆ , Molecules with an effective diameter <8 angstroms	Di-n-butylamine and larger	Aromatic hydrocarbon separation
13X	10	Powder 1/16-in. Pellets 1/8-in. Pellets	36 28.5 28.5	Molecules with an effective diameter <10 angstroms	Molecules with an effective diameter >10 angstroms, e.g. (C ₄ F ₉) ₃ N	Used commercially for general gas drying, air plant feed purification (simultaneous removal of H ₂ O and CO ₂) and liquid hydrocarbon and natural gas sweetening (H ₂ S and mercaptan removal).

For acid environments where the pH of the adsorbed water is below 5, AW300 and AW500 are used. These molecular sieves have the following properties.

Properties	AW300	AW500
Bulk Density, kg/m ³	888	728
Pellet Density, kg/m ³	1386	1165
Nominal Pore Size, °A	4	4
Avg. Heat of Adsorption, kJ/kg	3377	3377
Sp. Ht., kJ/kg·°C	0.63 at -51°C 0.80 at 38°C 1.00 at 238°C	

Desiccant Choice

The choice is primarily an economic exercise. The aluminas are the cheapest but require larger towers for a given water load, which increases capital cost and heat load. Molecular sieves are the most versatile but they are many times more expensive than gel or aluminas relative to their capacity for water. In effect, sieves have to be justified by factors like high inlet gas temperature, sour gases, outlet dewpoints that are below those of other desiccants and more selective separation requirements.

Any desiccant containing significant amounts of iron, like bauxite, is not very suitable for hydrocarbon streams that contain even trace (but measurable) amounts of sulfur compounds. Silica gel is a very suitable desiccant for use with low percentages of sulfur compounds. In any case where the pH of the adsorbed water will be 5 or less, an AW type sieve must be used.

Pressure Loss

The pressure drop across the entire unit normally is specified. Use a realistic value, not just some standard specification plucked from a file drawer. Unit cost is sensitive to pressure drop.

The pressure loss across the bed should be trivial compared to that across the unit. Most of it is in the piping manifold, switching valves and across controls. Allowing a higher pressure drop enables the designer to reduce the size of these components.

THE BASIC SYSTEM

Figure 19.1 shows the simplest dry desiccant system. It consists of two towers containing desiccant. One is drying while the other is regenerating.^(19.1) During regeneration all adsorbed materials are desorbed by heat to prepare the tower for its next cycle on-stream.

At the time shown, Tower 2 is drying. The main gas stream flows into the top of the tower and out the bottom. The filter shown is not used in all systems. As later discussions will detail, the regeneration cycle consists of two parts – heating and cooling. During the heating portion the regeneration gas is heated to 200-315°C [400-600°F]. The temperature depends on the desiccant being used and the character of the material being adsorbed.

The regeneration gas by-passes the heater to cool down the bed once the desiccant bed has been heated to the desired level. This cooling normally ceases when the bed is 10-15°C [18-27°F] higher than the inlet gas temperature.

The regeneration gas leaving the tower is cooled to condense the materials desorbed. After these are separated the gas usually returns to the main inlet gas stream. This regeneration gas will be 5-15% of the total throughput, with 10% being a good average. In gas dehydration, flow normally is always downward because of the higher allowable velocity in this direction. Upward regeneration is preferred even though it requires more valves and piping. Most bed contamination occurs at the top. By regenerating upward, the "steam" produced from the lower part of the bed helps remove the contamination. It can be removed without spreading throughout the bed. Cooling is optional. Upflow cooling saves two switching valves per tower (since unheated regeneration gas may be used), but requires dry gas. Downflow cooling (same direction as adsorption) is preferred if the cooling gas contains water.

There are three basic sources of regeneration gas in gas dehydration:

1. Inlet gas.
2. A closed cycle separate from the stream being dehydrated.
3. Dry effluent (tail) gas from the unit.

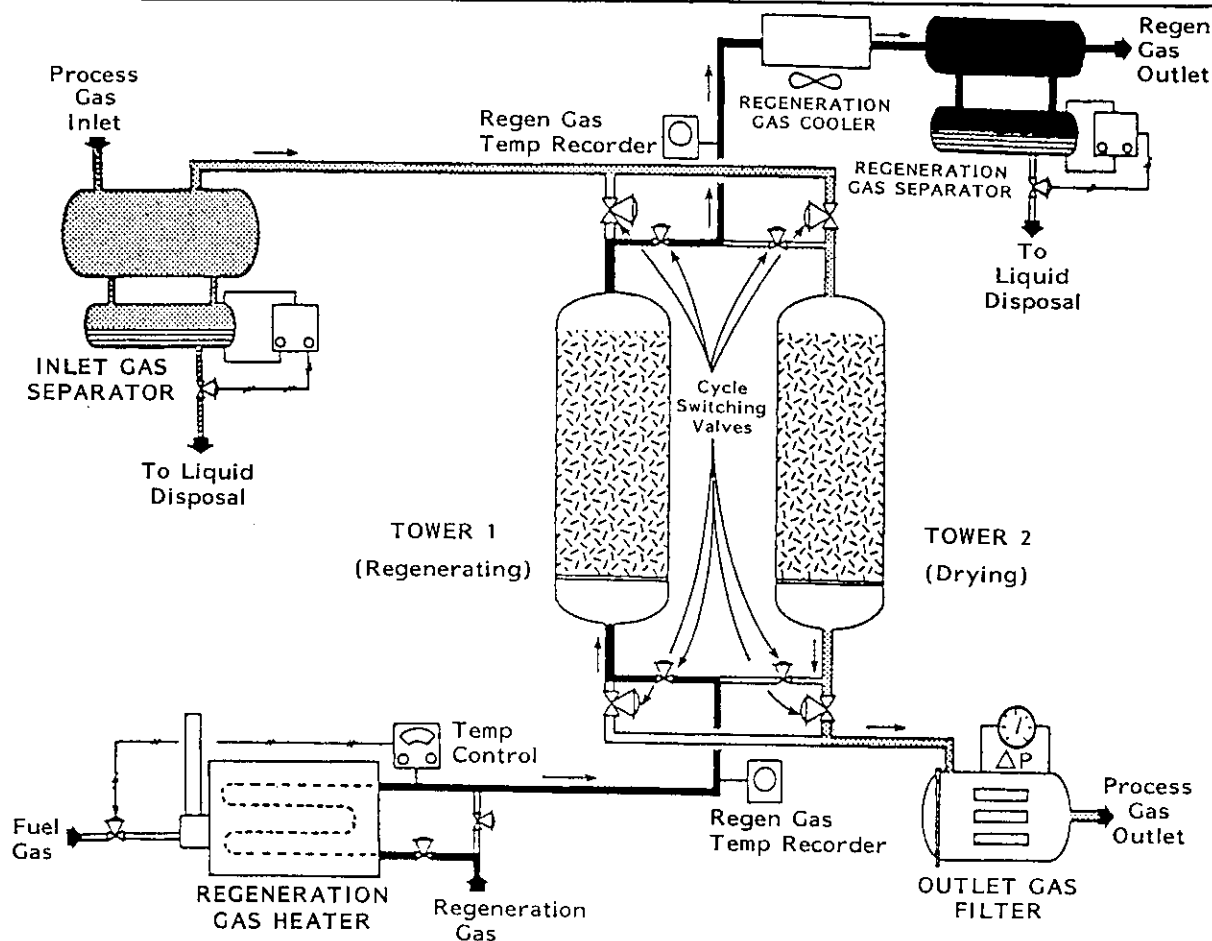
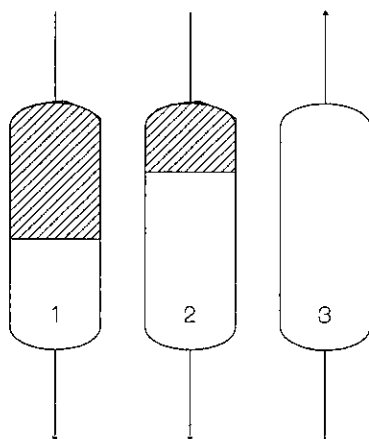


Figure 19.1 Flow Sheet of a Basic Two-Tower Dry Desiccant Unit

(1) involves some degree of re-saturation of the bed during cooling which limits the minimum water dewpoint achievable; (2) requires a separate piping system, and a compressor, and is not used very often; (3) is the most efficient, more costly than (1), but the norm in cryogenic drying service.

Most large dry desiccant units for natural gas drying contain more than two towers to optimize the economics. One design factor is the number of towers.

There are several ways one can use multiple towers. As shown in the illustration below using three towers, two are in-stream in parallel with the third being regenerated and cooled.



In this illustration, the shaded area inside towers 1 and 2 shows the progress of water adsorption in the bed or the portion of the bed which is essentially saturated with water. Below this area, the desiccant is capable of adsorbing more water. The bottom of this area represents the position of the adsorption front as it moves down through the bed with time.

The front in bed 1 is lower than in bed 2 because it has been on stream longer. When the leading edge of this front reaches the outlet, bed 1 will be switched to regeneration and beds 2 and 3 will be on

stream. Thus, at any one time, the two dehydrating towers possess different degrees of saturation. By the time bed 2 is ready for regeneration, bed 1 must be ready to go back on-stream.

The operating sequence of the towers on stream is:

1 and 2, 2 and 3, 1 and 3, 1 and 2, ad infinitum.

A similar arrangement could be used with four towers, with three on stream at a time. Obviously, the flow arrangement affects the cycle time chosen.

Figures 19.2 and 19.3 show a different use of a three-tower system as well as illustrating different regeneration/cooling arrangements.

Figure 19.2(a) uses by-passed wet gas for regeneration in what is called an open cycle. A portion of this wet gas is by-passed and goes to the tower which has been regenerated but is ready for cooling. The bed being cooled pre-heats the gas to the heater to save on fuel. The gas leaves the heater at a temperature suitable for regeneration and enters the top of the wet tower ready for regeneration. The gas leaving flows to a condenser-cooler. Liquids formed are separated and the gas returns to the inlet main gas stream. This cooling/heating process uses the least amount of equipment, gives maximum heat efficiency and minimizes pressure drop. However, the condensation efficiency is not as high as for some other arrangements.

In Figure 19.2(b) the regeneration gas is heated, enters the regenerating tower, is cooled and then enters the tower requiring cooling. The warmed gas leaving the tower exchanges heat with the entering regeneration gas and then returns to the main stream. This heating/cooling system has a better condensation efficiency than cooling/heating. However, it requires an extra heat exchanger. It also pre-saturates the bed being cooled more than the cooling/heating system.

Figure 19.3(a) shows the use of inlet gas for regenerating and the effluent gas for cooling. In this system complete regeneration must be achieved before cooling a bed or desorbed components can enter the plant outlet. It requires more equipment than any of the other processes and uses more regeneration gas. This must be justified by good condensation efficiency and reduced pre-saturation. The latter may be particularly important if recovery of the lighter hydrocarbons is desired.

Figure 19.3(b) uses closed cycle regeneration with effluent gas cooling. Closed cycle regeneration offers the advantage of very good condensation efficiency. It has been found that placing the tower to be cooled in the closed cycle is an unsatisfactory practice. The closed cycle obviously is more complex and more expensive but it may sometimes be justified for hydrocarbon dewpoint control.

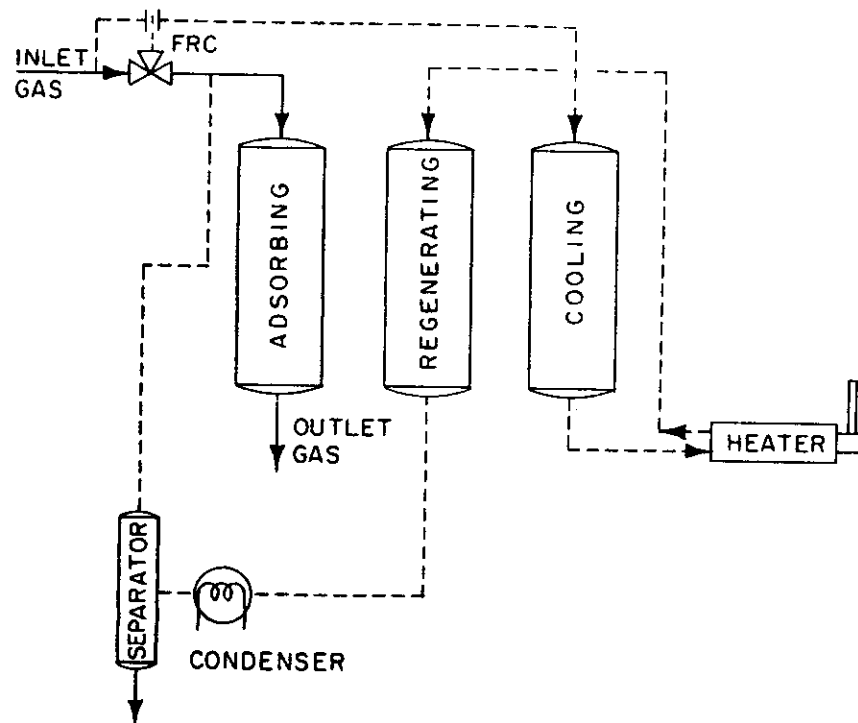
Shown on page 376 is a piping manifold for a three tower plant using a general configuration like that shown in Figure 19.2 and 19.3. Two-way valves are shown. Three-way valves are sometimes used, but are not recommended. This illustrates the basic type of layout required. It is important that the valves not leak.

All of these systems shown regenerate at essentially full line pressure, using hot gas. In some small units it may be feasible to use heating coils imbedded in the bed. In this case, enough carrier gas must be used to carry out the water vapor formed.

One can also reduce tower pressure to regenerate. Some air driers use this approach, which requires no heater. Carrier gas is needed to once again carry out the water vapor. One must reduce pressure slowly enough to eliminate desiccant breakage. This approach has not been used, so far as I know, for large natural gas driers.

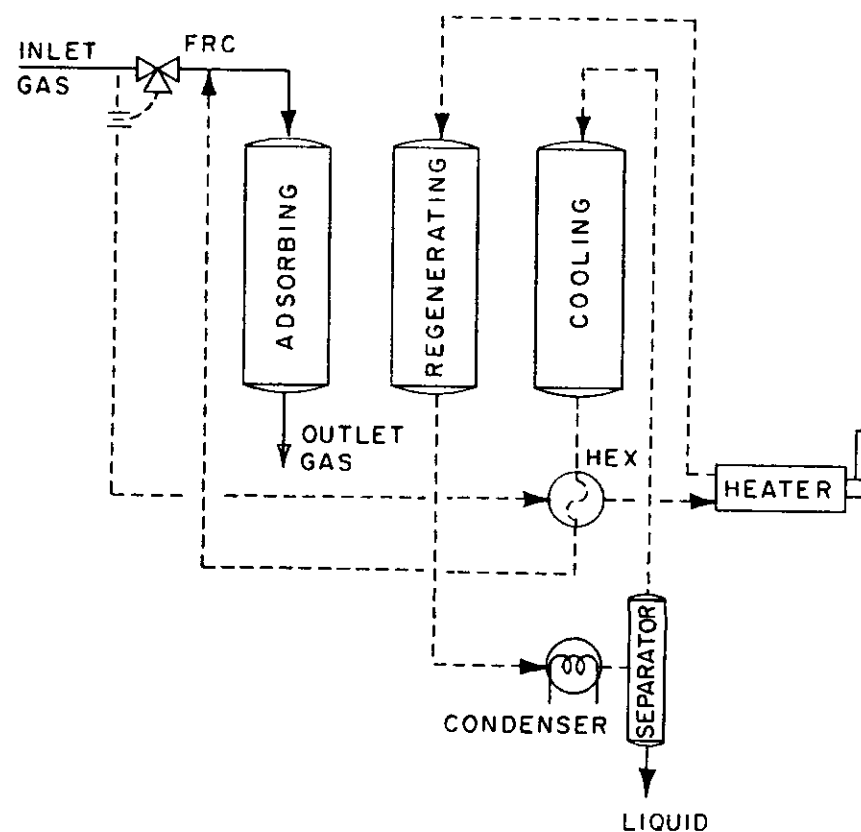
One of the major operating costs in dry desiccant plant is for heater fuel. If it is available, waste heat may be used. Another improvement over traditional efficiency may be obtained by use of better insulations of towers, piping and valves.

A more detailed discussion of desiccant plants and their operations is available in Reference 19.1.



(a)

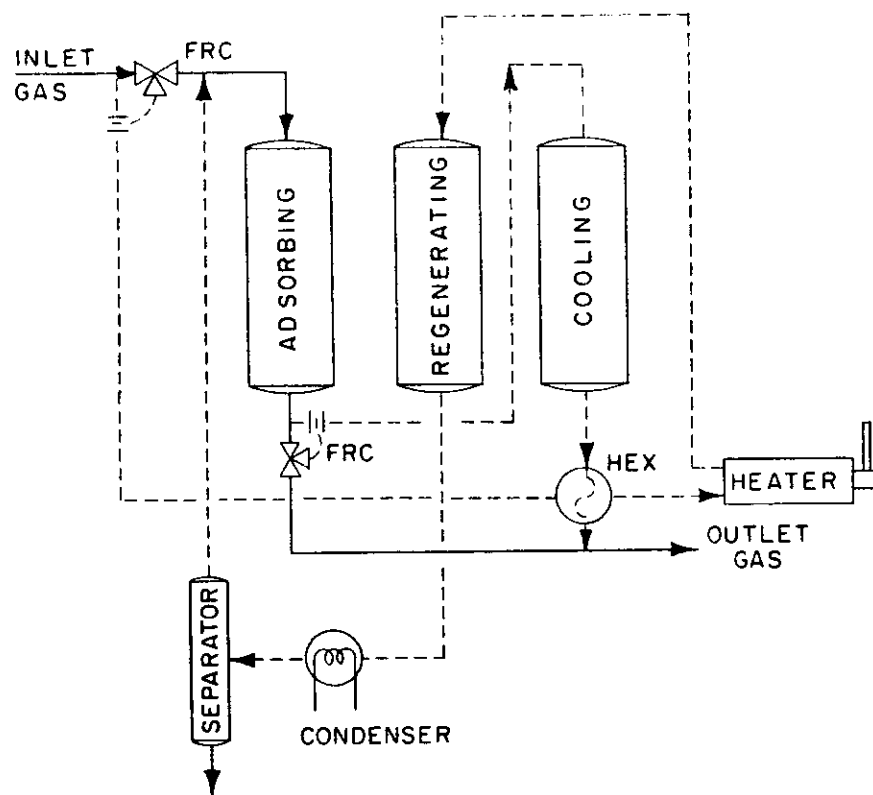
Schematic view of a typical three-tower plant using open cycle wet gas cooling and heating on regeneration in that order



(b)

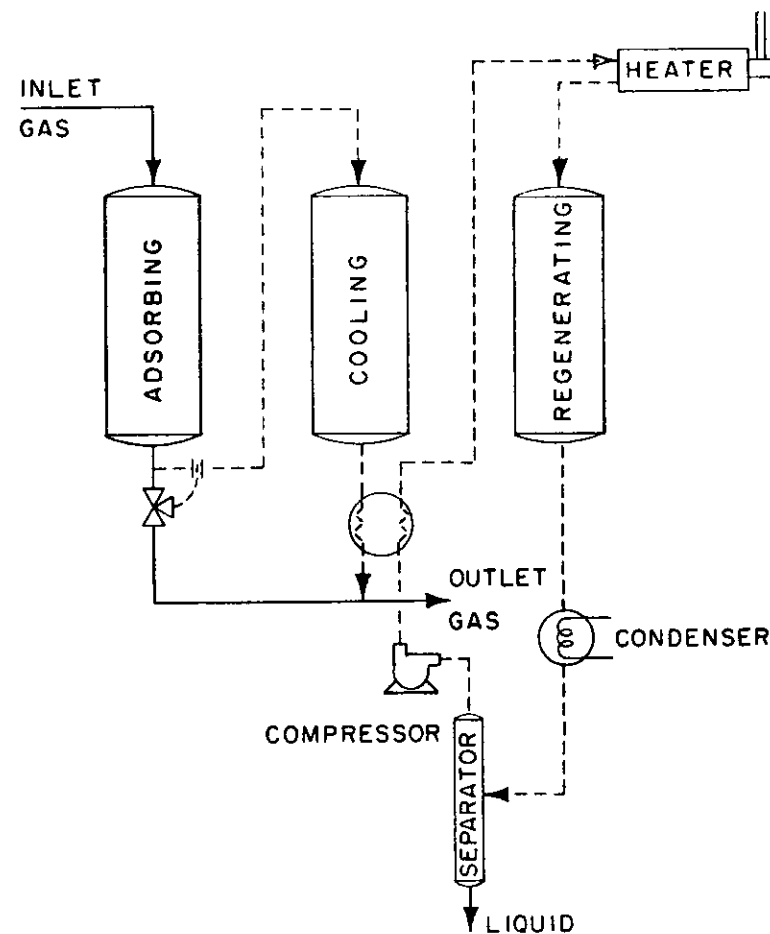
Schematic view showing open cycle wet gas heating and cooling on regeneration in that order

Figure 19.2 Two Possible Configurations of a Three-Tower Plant



(a)

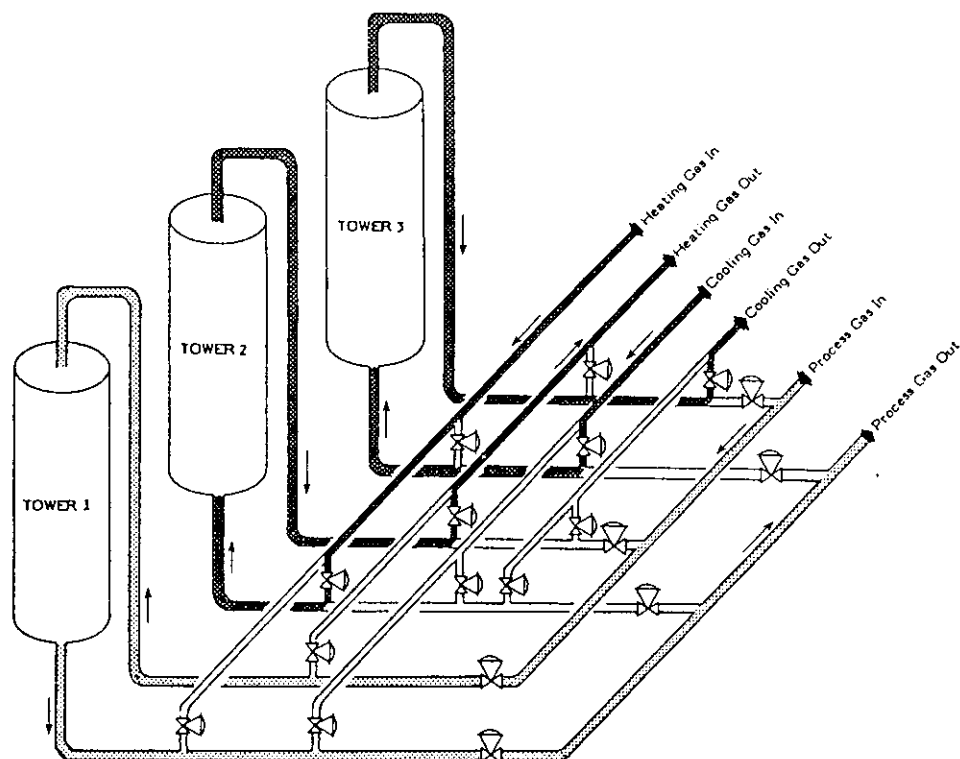
Schematic view showing open cycle wet gas heating and effluent gas cooling



(b)

Schematic view of closed cycle regeneration
using effluent gas cooling

Figure 19.3 Two Further Possible Configurations of a Three-Tower Plant



Pipe Manifold for 3-Tower Adsorber Plant
Tower 1 is Adsorbing, Tower 2 is Heating, and Tower 3 is Cooling

THE REGENERATION CYCLE

Figure 19.4 is a temperature-time plot of the type obtained from a dry desiccant plant. The specific plot is for a two-tower plant of the type shown in Figure 19.1 when using an 8-hour cycle.

At first, the entering gas loses its heat primarily to "heat up" the vessel contents (plus the vessel itself if no internal insulation is used). At about 120°C [248°F], the water will start vaporizing and the curve flattens, while most of the heat input goes into such vaporization. It is often assumed in design that all of the water desorbs at an average temperature of about 125°C [257°F], which is designated by point T_B .

Following desorption, the outlet bed temperature starts rising again. The process of heating is terminated when the desired outlet temperature is reached. In the simplest plants, the heater is simply by-passed and cool gas enters the bed to prepare it for switching to drying service. In this scheme, the gas used is near saturation. Pre-saturation of the bed with water will occur as the bed cools. Cooling should be terminated at 50-55°C [122-131°F] to minimize this problem. In more complex plants dry exit gas or some outside source is used for cooling to minimize pre-saturation. In low temperature plants, one of the dry, process streams may be used. The bed is used as a kind of regenerative heat exchanger.

When using cycle lengths above 4 hours, good regeneration usually can be obtained with a maximum exit gas temperature of 180-205°C [350-400°F] for silica gel and 275-300°C [530-570°F] for molecular sieve. The final heating is necessary to remove the "heel" from the bed – the heavy hydrocarbons and contaminants that do not vaporize at lower temperatures. The minimum temperature possible should be used to minimize heat load and fuel consumption. A good operator will find this optimum temperature by trial and error.

The nomenclature used in this figure will be used later when outlining a regeneration cycle calculation. Although Figure 19.4 is for an 8-hour cycle, the relative time for heating and cooling is indicative of any cycle above 4 hours in a gas dehydrator.

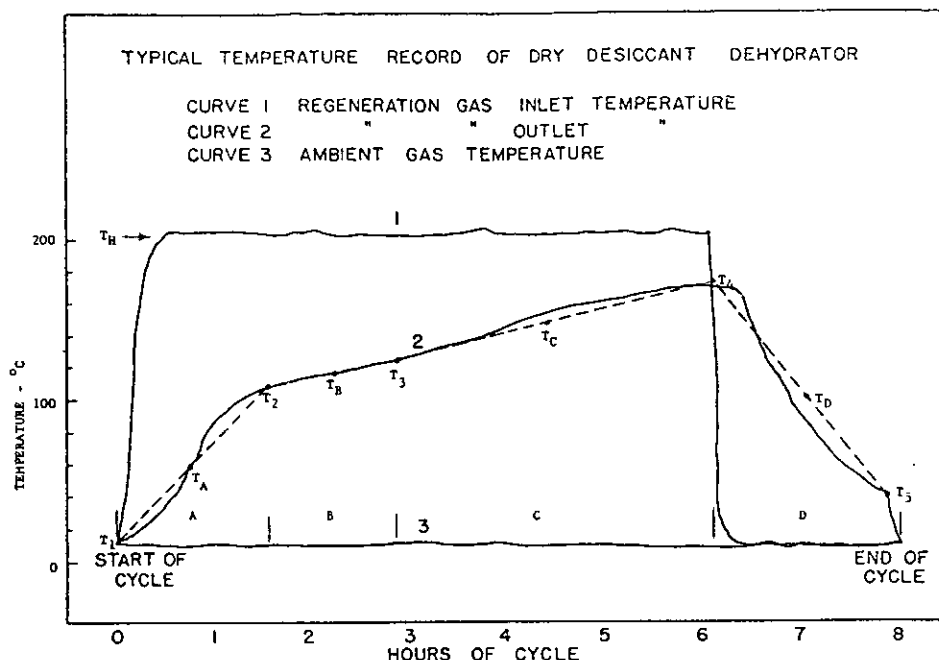


Figure 19.4 Temperature Curves for a Two-Tower Plant

Table 19.2 summarizes the characteristics and common regeneration practices in gas dehydration.

THE NATURE OF ADSORPTION

Figure 19.5(a) illustrates the basic behavior of an adsorbent bed in gas dehydration service. During normal operation in the drying (adsorbing) cycle, three separate zones exist in the bed: 1) equilibrium zone, 2) mass transfer zone (MTZ), and 3) active zone.

In the equilibrium zone the desiccant is saturated with water. It has reached its equilibrium water capacity based on inlet gas conditions and has no further capacity to adsorb water.

Virtually all of the mass transfer takes place in the MTZ. A concentration gradient exists across the MTZ. This is illustrated in Figure 19.5(b) for various times throughout the cycle. Curves 1-3 show the formation of the MTZ; curve 4 reflects the concentration gradient for the MTZ position in Figure 19.5(a). Curve 6 shows the concentration gradient at breakthrough. Notice the adsorbate bed saturation is 0% at the leading edge of the MTZ and 100% at the trailing edge.

The third zone is the active zone. In the active zone the desiccant has its full capacity for water and contains only that amount of residual water left from the regeneration cycle.

When the leading edge of the MTZ reaches the end of the bed, breakthrough occurs. If the adsorption process is allowed to continue, the water content of the outlet gas will increase following the traditional "S" curve. Breakthrough curves are illustrated in Figure 19.5(c) for three MTZ lengths.

Figure 19.5(d) shows the location of MTZ's in multicomponent adsorption typical of hydrocarbon and water adsorption on silica gel. As the gas enters a dry desiccant bed, all of the adsorbable components are adsorbed at different rates. After the process has proceeded for a very short period of time, a series of adsorption zones will appear. These zones represent the length of tower involved in the adsorption of any component. Behind the zone all of that component entering has been adsorbed on the bed. Ahead of the zone, the concentration of that compound is zero (unless some is left from a previous adsorption or on regeneration). These zones form and move down through the desiccant bed. Water would be the last zone

TABLE 19.2

Summary of the Operating Characteristics Dry Desiccant Dehydration

RECOMMENDED OPERATING RANGE:

At temperatures preferably below 50°C in order to utilize the higher desiccant capacity. The gas should enter the unit above its hydrate point. There seem to be no effective pressure limitations.

DESICCANT SERVICE:

Normally from three-five years in the absence of poisoning. Service limited by loss of capacity, dusting and breakage.

DEHYDRATION OBTAINED:

Essentially bone dry gas produced. Most units will give a -75 to -125°C [-103 to -193°F] dewpoint during the first part of the cycle.

LENGTH OF CYCLE:

Varies with water loading and gas rate, usually 8-24 hours.

REGENERATION:

Final regeneration temperature varies with type of desiccant, usually 175-300°C [347-572°F]. About 5 to 15 percent of the total gas stream used for regeneration. Normal eight hour cycle uses about six hours for heating, two hours for cooling in two tower plant.

ADVANTAGES:

1) Low exit dewpoint obtained. 2) Gives effective dewpoint depressions over a wide range of operating conditions. 3) Compact, particularly the smaller units. 4) Relatively low initial investment for small amounts of gas, such as for instruments, where "batch" or "semi-batch" operation is satisfactory. 5) Rated capacity may be increased if some wet gas is by-passed around the unit and recombines with dry gas, where the unit gives an exit dewpoint lower than that needed or specified.

DISADVANTAGES:

1) High initial investment. 2) Desiccant sensitive to poisoning, particularly from heavy oils, which frequently requires expensive desiccant replacement. 3) Rated capacity of the unit declines with pressure. 4) Pressure drop is higher than with liquid desiccant systems. In compressed gas systems this increases the required compressor horsepower. 5) At flow rates below the rated capacity, the regeneration heat load is high in relation to the amount of gas processed.

Summary of Regeneration Practices

METHOD OF HEATING:

Usually part of the main stream is by-passed which is either returned to the system downstream or is recombined and dehydrated. Liquefied petroleum gases which have been vaporized or super-heated steam are also used.

REQUIRED TEMPERATURE:

In most instances 175-300°C [347-572°F] is recommended. Higher temperatures increase capacity but shorten effective life of desiccant. Majority of water is driven off at 125°C [257°F]. Gas temperature from heater usually 200-350°C [392-662°F].

DIRECTION OF GAS FLOW:

Usually countercurrent to main flow in vertical towers in long cycle units. Concurrent in short cycle units.

GAS FLOW RATE:

About 5 to 15 percent of main gas stream. Percentage varies with main gas flow rate and with design of unit. Flow rate must be sufficient to supply the necessary heat in time allotted, for regeneration.

WATER REMOVAL:

Where regeneration gas recombined with main gas, water from regeneration is removed by cooling with main gas stream or by water cooler. Maximum heat load occurs when bed reaches about 125°C [257°F], at which time most of water is coming off. Heat exchanger should be designed for this condition.

HEAT REQUIRED:

Heat of desorption of water plus sensible heat to bring water vapor, desiccant and container up to temperature.

TIME REQUIRED:

To bring exit regeneration gas up to desired temperature. Usually about 65-75 percent of total cycle time.

PRESSURE:

Full line pressure satisfactory.

increase velocity, 10-20
- better sweep for the bed
- assume degassing when regeneration -

water for cleanup
in switching
or bit in product gas

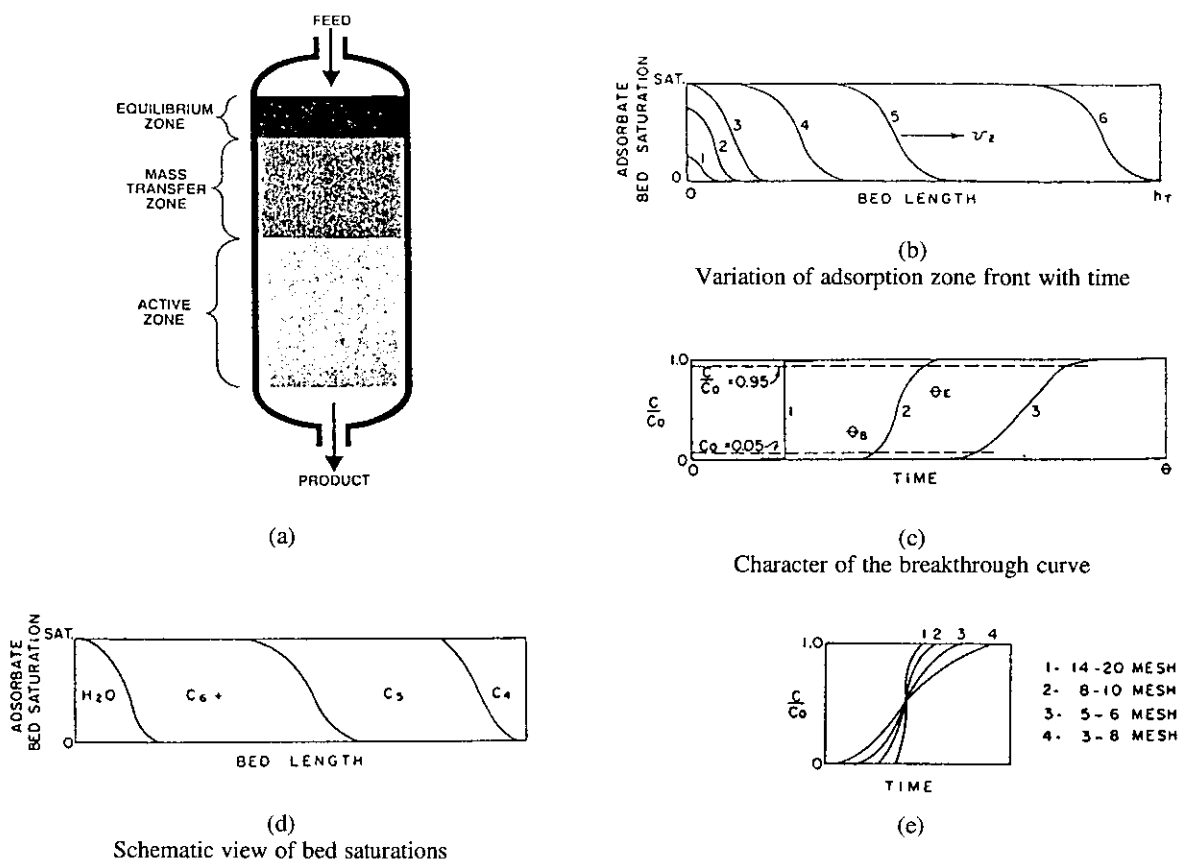


Figure 19.5 Schematic Portrayal of Adsorption Process

formed. On all materials except carbon it will displace the hydrocarbons if enough time is allowed to do so. If molecular sieve is used, adsorption of the C_4 - C_6 fractions will not occur because these molecules cannot fit in the desiccant structure.

100% of any component is adsorbed on the desiccant until the front of its zone reaches the outlet of the bed. When the back of its zone reaches the outlet of the bed, no more adsorption of that component will occur. It will furthermore be displaced almost entirely by the component in the zone following it down the bed if the cycle is continued. If the process continues long enough, no effective amount will remain on the bed.

With silica gel, at commercial flow rates and tower configurations usually employed, pentane will have a breakthrough time of from 12-20 minutes. Methane and ethane break out almost instantaneously. If the process cycle proceeds beyond 30-40 minutes, all but the heaviest hydrocarbons will have been displaced out of the bed. From this time on, primarily dehydration is taking place. thus, the performance of a given unit is dependent on the cycle length used.

For very short cycles both hydrocarbon adsorption and dehydration occur. The hydrocarbon recovery will substantially reduce dehydration costs charged to the unit or eliminate them. Such units may be used to simultaneously control water and hydrocarbon dewpoints at minimum cost. Their potential application is far greater than their use to date would indicate.

Part (e) shows the effect of desiccant size on the length of the zone. The steeper the zone, the sharper the separation, the better the process. Therefore, the desiccant used should always be the smallest compatible with the drop limitations. The smallest size will seldom be greater than 14 mesh (Tyler Screen Scale) in most commercial natural gas installations.

Other factors which affect the length of the MTZ include gas velocity – increasing velocity increases length – contaminants, water content and relative saturation of the inlet gas. Contaminants are particularly insidious because they can slow the mass transfer process (lengthen MTZ) by providing additional resistance.

The length of the MTZ has a significant effect on the useful capacity of the desiccant since the MTZ is left in the bed at the end of the adsorption cycle. Remember, the desiccant in the MTZ is only partially saturated with water!

THE PROCESS VARIABLES

The first step in planning and specification is to establish the waterload per 24 hours based on gas flow rate, P, T and water content. Also needed is the desired outlet water dewpoint (or water content). The unit will be sized by assuming all inlet water is removed regardless of the outlet dewpoint specified. All commercial desiccants are capable of producing water dewpoints below -60°C [-76°F].

The actual outlet dewpoint will depend on the desiccant chosen and design of the unit. With designs used to date, the following dewpoints are achievable:

Desiccant	Outlet Dewpoint
Alumina	-73°C [-100°F]
Silica Gel	-60°C [-76°F]
Molecular Sieves	-90°C [-130°F]

Slightly lower dewpoints than these have been reported in some instances.

To achieve these low dewpoints, the drying cycle must be such that the MTZ never reaches the end of the bed. So-called *short cycle* units are sometimes used in this application. Heavier hydrocarbons are recovered as well as producing very low exit water dewpoints.

Once the water load is fixed it is simply necessary to expose the gas to enough active desiccant to adsorb it. There are an infinite number of cycle length-number of tower combinations to accomplish this. Which combination is optimum?

Optimization considerations include: (1) Cycle time, (2) Allowable gas flow rate, (3) Desiccant capacity, (4) Required outlet water dewpoint, (5) Total amount of water to be removed, (6) Dynamic adsorption performance of the desiccant tower, (7) Regeneration requirements and (8) Pressure drop limitations. These are not independent variables. Consequently, some have to be fixed for the purpose of a calculation. True optimization will require repeating the calculation several times to obtain the best combination of variables.

In making these calculations one must assume a particular desiccant in order to have capacities and physical properties for use in the calculation.

Desiccant Capacity

The capacity of a desiccant for water is expressed normally in mass of water adsorbed per mass of desiccant. There are three capacity terms used.

Static Equilibrium Capacity – the water capacity of new, virgin desiccant as determined in an equilibrium cell with no fluid flow.

Dynamic Equilibrium Capacity – the water capacity of new, virgin desiccant where the fluid is flowing through the desiccant at a commercial rate.

Useful Capacity – the design capacity that recognizes loss of desiccant capacity with time as determined by experience and economic considerations and the fact that all of the desiccant bed never can be fully utilized.

The static equilibrium capacity has no direct use in design although it shows the effect of P, T and gas composition on capacity. As later calculations will illustrate, dynamic and useful capacity are used directly in calculations. Dynamic capacity typically is 40-60% of static capacity.

All desiccants degrade in service. Figure 19.6 is a typical curve for silica gel. Other desiccants will have the same shaped curves in normal service, although the values will vary.

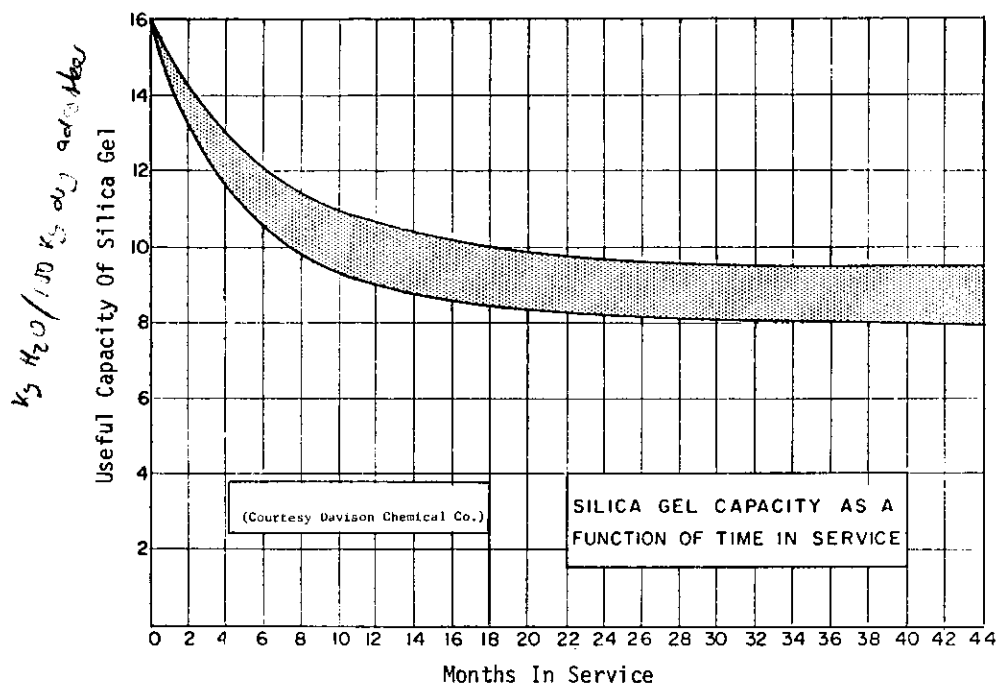


Figure 19.6 Silica Gel Capacity as a Function of Time in Service

Normal degradation occurs through loss of effective surface area on repeated regeneration. This loss is rapid at first and then becomes more gradual as the desiccant "matures." Abnormal degradation occurs primarily through blockage of the small capillary or lattice openings which control access to the interior surface area. Heavy oils, amines, glycols, corrosion inhibitors and the like, which cannot be removed by regeneration, can reduce the capacity to uneconomic levels in short periods of time. There is no room for wishful thinking. If these contaminants are present ahead of the unit, provision to handle them must be made just ahead of the unit. An ordinary separator will help, but filter separation or a "guard bed" of spent desiccant or some packing material is usually desirable.

Water is always a problem. Salt water entering will evaporate and fill the bed with salt. With the gels this water will cause bead breakage unless a guard section is provided at the inlet. The only good solution is to let no liquid water enter the bed.

The useful capacity used should be such that economic desiccant life will be obtained. For normal service the following are commonly specified as useful capacity:

Bauxite - 4-6 kg water per 100 kg of desiccant.	Gels - 7-9 kg water per 100 kg of desiccant.
Alumina - 4-7 kg water per 100 kg of desiccant.	Molecular Sieves - 9-12 kg water per 100 kg of desiccant.

In normal service, outlet dewpoints will be below -60°C [-76°F] for silica gel and -90°C [-130°F] for mol sieve until water zone breakthrough. Where lower dewpoints are needed, as in cryogenic plants, the overall effective capacity might be lower because the adsorptive driving force is so low at the exit end of the bed.

When applying the foregoing design capacities to water removal in natural gas service, it is customary to assume that the gas enters saturated and *all* water will be removed. This introduces a slight safety factor in most cases. Most natural gas is saturated as it leaves the reservoir, but subsequent pressure and temperature changes alter this. The relative saturation (RS) of the entering gas has a calculable effect on the desiccant performance, particularly silica gel and alumina.

4A and 5A molecular sieves tend to degrade slower because their pore size is such that heavy hydrocarbon molecules are excluded from the interior. However, a heavy external coating will still compromise their performance.

Most of the contamination occurs near the inlet. Sometimes the desiccant tower is designed so that this portion of the bed may be replaced more frequently than the total bed.

A form of degradation can occur if liquid water enters the bed. Some desiccants explode in the presence of liquid water. The *finer* thus produced increase pressure drop, reduce effective capacity, and may get into the pipeline and cause damage to rotating machinery downstream. A layer of water resistant desiccant may thus be placed on top of the bed to minimize this problem. The most positive solution is effective inlet scrubbing.

For pipeline service where outlet dewpoints are relatively high, it is desirable to continue the cycle until the water front reaches the end of the bed. Upon regeneration the steam formed helps clean the desiccant. Thus, cycle length may be variable with desiccant age and gas flow rate. For this reason, outlet dewpoint control of cycle time is preferable to mere time cycle control although the latter is simpler and cheaper. At worst, a dewpoint recorder should be placed on the outlet so that one can adjust performance or obtain critical data for troubleshooting functions.

ADSORBER SIZING EQUATIONS

For a given set of gas flow conditions, water loading, cycle length and tower configuration, one can size the desiccant bed. It must have enough area to meet velocity limitations, enough mass (surface) to hold the water adsorbed, and possess enough length so that the MTZ has not passed through the desiccant bed in the cycle length chosen.

The true dynamics of the adsorption process are difficult to simulate. The calculation procedure that follows is, however, a good approximation for estimation of dehydration behavior.

Desiccant Capacity

$$(x)(h_B) = (x_s)(h_B) - (0.45)(h_Z)(x_s) \quad (19.1)$$

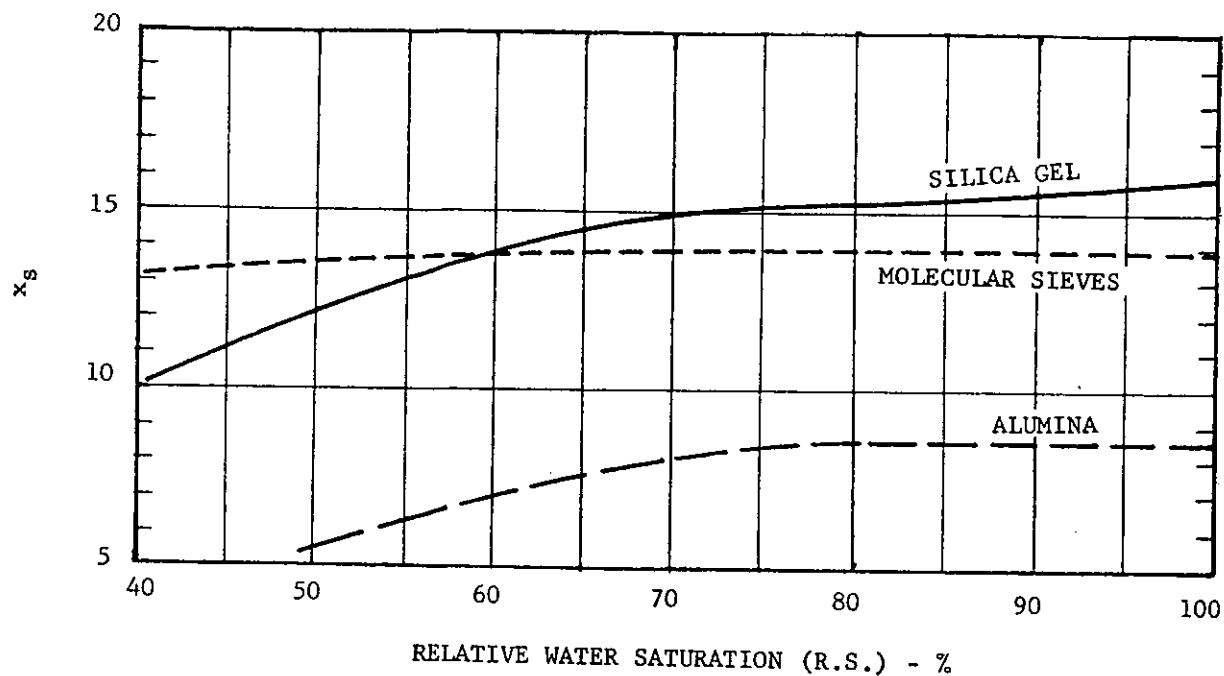
Where:

- x = maximum desiccant useful capacity (U.C.), kg water per 100 kg desiccant
- x_s = dynamic capacity at saturation (U.C.), kg water per 100 kg desiccant
- h_Z = MTZ length
- h_B = bed length (or length of bed to front of adsorption zone)

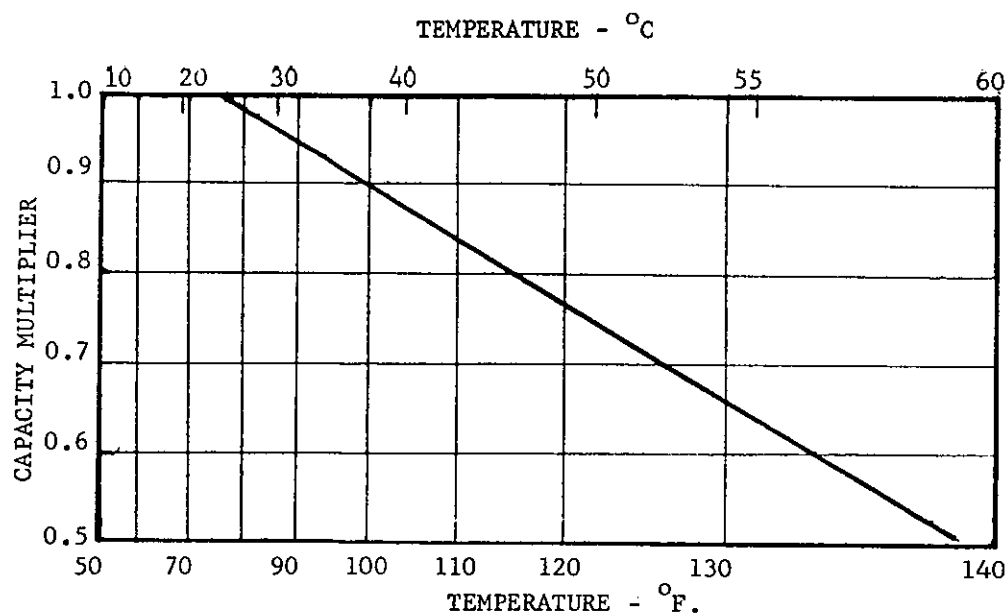
(Values of x 's and h 's may be in metric or English units so long as consistent).

The dynamic capacity " x_s " must reflect desiccant condition and other such factors. It is the effective capacity of the desiccant -- for water -- *behind* the adsorption zone. Since desiccant degrades in service, the value used must reflect a capacity at some future time to optimize desiccant replacement cost.

Figure 19.7(a) may be used to predict " x_s " in Equation 19.1 as a function of relative saturation. Remember ... the relative gas saturation is fixed by the last pressure and temperature at which liquid water was present -- the last separator producing any water or the reservoir. If any water is drained off the inlet



(a)



(b)

Figure 19.7 Effect of Relative Saturation and Temperature on the Dynamic Capacity of Desiccants

scrubber, the inlet gas is saturated. If not, the *saturated* water content at the previous contact divided by that at the dehydrator give the R.S. value as a *fraction*.

The values in Figure 19.7(a) are lower than the theoretical numbers published for air. They are for natural gas and reflect the competition of the hydrocarbons for adsorption surface and expected dynamic saturation after a short time use in an actual plant – with normal degradation only.

For *gels* and *aluminas*, the values in Figure 19.7(a) must be corrected for temperature. No temperature correction is needed for *molecular* sieves in the temperature range shown. The value from Figure 19.7(b) is multiplied by that in Figure 19.7(a) to obtain the "x_s" for use in Equation 19.1.

The "x" obtained from Equation 19.1 will be the useful capacity of virgin, activated desiccant. This is greater than the useful capacity after degradation as shown in Figure 19.6. In effect, Equation 19.1 simply corrects for that amount of bed unused because of zone length.

MTZ Length

The MTZ length depends on gas analysis, gas flow rate, relative saturation of the water in the gas and the loading capability of the desiccant. Pressure has only a small effect, particularly above 2.1 MPa [300 psia].

The numerical value of 0.45 in Equation 19.1 is an average number based on test. It is a function of MTZ but only varies from 0.40-0.52 in a wide range of applications. The value used is the mode of the distribution curve for most services.

For silica gel, the MTZ length may be estimated from the equation

$$h_Z = A \left[\frac{q^{0.7895}}{v_g^{0.5506} (R.S.)^{0.2646}} \right] \quad (19.2)$$

Where: A = constant
 h_Z = MTZ length
 q = water loading
 v_g = superficial velocity
 R.S. = percent relative saturation of inlet gas

Metric	English
141	375
cm	in.
kg/h·m ²	lb/(hr·ft ²)
m/min	ft/min
	dimensionless

(The superficial velocity is based on bed diameter)

The values of "h_Z" from Equation 19.2 are those quoted from Reference 19.3 for air drying. The values, however, are useful for natural gas drying. Reference 19.8 represents a slightly more complex approach.

In natural gas service, when using Equation 19.2, the following multipliers are suggested for alumina and molecular sieves:

Alumina – 0.8 times h_Z for gel

Molecular Sieve – 0.6 times h_Z for gel

Shorter zones are obtained with these materials because they have less capacity for hydrocarbon. As noted in Table 19.1, 3A or 4A sieve will adsorb no heavier hydrocarbons.

Water Loading

$$q = 0.053 \left[\frac{(\text{Flowrate}) (W)}{d^2} \right] \quad (19.3)$$

Where: q = water loading
 Flowrate = gas rate
 d = bed diameter
 W = water content

Metric	English
kg/h·m ²	lb/hr-ft ²
10 ⁶ std m ³ /d	MMscf/d
m	ft
kg/10 ⁶ std m ³	lb/MMscf

This is the water loading on a mass basis. Equation 19.3 is merely a conversion from water content per standard volume divided by bed cross-sectional area.

Breakthrough Time

$$\theta_B = \frac{(0.01) (x) (\rho_B) (h_B)}{q} \quad (19.4)$$

$$h_B = \frac{(127.3) (\text{wt of water adsorbed/cycle})}{(\rho_B) (d)^2 (x)} \quad (19.5)$$

Where: wt water is expressed in
 q_B = breakthrough time
 ρ_B = bulk density of desiccant
 h_B = bed length
 q = water loading

Metric	English
kg	lb
h	hr
kg/m ³	lb/ft ³
m	ft
kg/(h·m ²)	lb/(hr-ft ²)

General Conversion Equations

$$w = \frac{(A) (v_g) (\gamma_g) (P)}{(T) (z)} \quad (19.6)$$

$$w = \frac{(B) (\text{Flowrate}) (\text{MW Gas})}{d^2} \quad (19.7)$$

$$d = \left[\frac{(C) (\text{Flowrate}) (z) (T)}{(P) (v_g)} \right]^{0.5} \quad (19.8)$$

$$v_g = \frac{(C) (\text{Flowrate}) (z) (T)}{(P) (d^2)} \quad (19.9)$$

$$q = \frac{(E) (W) (P) (v_g)}{(T) (z)} \quad (19.10)$$

		Metric	English
Where:	A = constant	209	162
	B = constant	2214	140
	C = constant	307	25
	E = constant	0000 173	0.002 16
	w = gas mass velocity	kg/(h·m ²)	lb/(hr·ft ²)
	v _g = superficial gas velocity	m/min	ft/min
	d = bed diameter	m	ft
	P = adsorber pressure	kPa	psia
	T = inlet gas temperature	K	°R
	W = water content	kg/10 ⁶ std m ³	lb/MMscf
	q = water loading	kg/(h·m ²)	lb/(hr·ft ²)
Flowrate =		10 ⁶ std m ³ /d	MMscf/d
	z = compressibility factor	dimensionless	
	g _g = gas relative density	dimensionless	

Equation 19.10 is an algebraic combination of Equations 19.3 and 19.8.

Allowable Gas Flow Rate (Bed Area)

The main flow is downward through the bed, and high superficial velocities (based on bed cross-sectional area) may be used. Most designers, however, find that less "fines" and desiccant breakage occur if the rate does not exceed 1.0 to 1.5 times the maximum upflow velocity predicted by Ledoux^(19.4) to prevent churning. Modifying his equation for downflow,

$$v_g = C \left(\frac{D_p}{\rho_g} \right)^{0.5} \quad (19.11)$$

		Metric	English
Where:	v _g = gas superficial velocity	m/min	ft/min
	ρ _g = gas density	kg/m ³	lb/ft ³
	D _p = particle diameter (average)	m	ft
	C = constant	1200	540

The particle diameter, D_p, is found from the mesh size of the desiccant used. Alumina and bauxite are granular materials; the gels are somewhat spherical. Their size is determined by screening through a series of screens bearing a *mesh size*. Different scales are used. The most common is the Tyler Screen Scale. The table below shows the common mesh sizes used for most desiccants.

Tyler Mesh	Screen Opening	
	mm	inches
3	6.680	0.263
4	4.699	0.185
5	3.962	0.156
6	3.327	0.131
7	2.794	0.110
8	2.362	0.093
9	1.981	0.078
10	1.651	0.065
12	1.397	0.055
14	1.168	0.046

The U.S. scale is very similar to the Tyler Scale. A typical gel will have a size like 3-8 mesh or 4-8 mesh. The first number is the size of screen all particles pass through; the second number is the size opening all particles are retained on. The size distribution is never uniform but for calculation purposes an average size for the range may be used for D_p .

Molecular sieves are supplied in spherical and in pellet form, as well as a powder. Pellets 1.59 mm and 3.18 mm [1/16 and 1/8 in.] are available in most grades. Comparable sphere sizes are marketed by some vendors.

Equation 19.11 is conservative for gels and sieves which tend to produce less "fines" than granular materials with sharp edges. For a typical 4-8 mesh gel this equation produces the following superficial velocities at a temperature of 27°C.

Pressure		Gas Velocity	
MPa	psia	m/min	ft/min
2.6	400	12-16	40-54
3.4	500	11-15	37-49
4.1	600	10-13	33-44
4.8	700	9-13	31-41
5.5	800	8-12	27-38
6.2	900	8-11	26-35
6.9	1000	8-10	25-33
7.6	1100	7-10	24-32
8.3	1200	7-9	23-31

Velocities up to 18 m/min [60 ft/min] have been used successfully in some installations. The velocity affects both diameter and length of the bed. As it increases, diameter decreases and length increases. The length requirement for a given diameter is fixed by two factors – total desiccant needed to handle the water load and the effect of v_g on zone front velocity. Cycle length must be adjusted for given tower dimensions to satisfy both criteria.

An alternative method for determining superficial velocity in a mol sieve bed uses the method of Ergun^(19.14) which relates ΔP to v_g , μ , ρ and desiccant size.

$$\frac{\Delta P}{L} = B \mu v_g + C \rho_g v_g^2 \quad (19.12)$$

Where: $\Delta P/L$ = pressure drop/length
 μ = gas viscosity
 ρ_g = gas density
 v_g = superficial gas velocity

Metric	English
kPa/m	psi/ft
cp	cp
kg/m ³	lbm/ft ³
m/min	ft/min

Constants for Equation 19.12 are:

Particle Type	Metric		English	
	B	C	B	C
1/8" bead	4.16	0.00135	0.0560	0.0000889
1/8" extrudate	5.36	0.00189	0.0722	0.000124
1/16" bead	11.3	0.00207	0.152	0.000136
1/16" extrudate	17.7	0.00319	0.238	0.000210

Most designs are based on a $\Delta P/L$ of about 7-10 kPa/m [0.31-0.44 psi/ft].

ADSORBER SIZING CALCULATION

The equations in the previous section serve as the basis for a calculation based on a given *cycle length*, *number of vessels and their configuration*, and a given desiccant.

1. Calculate the total water load for the cycle length for which the ensuing calculation applies.
2. Divide (1) by the number of towers on stream in parallel to find water load per cycle per tower.
3. Establish useful capacity "x" from Equations 19.1 and 19.2, and Figure 19.7, or by a company standard.
4. Multiply water load per cycle per tower by the number of cycles a tower operates before regeneration and divide the result by "x" to find the *total mass* of desiccant needed per tower.
5. Divide mass in (4) by desiccant bulk density to find *volume* of desiccant required per tower.
6. Calculate minimum bed diameter by one of two methods:
 - a. Calculate allowable superficial gas velocity from Equation 19.11, 19.12, or company specification.
7. Determine breakthrough time from Equation 19.4.
8. Determine minimum bed length from Equation 19.5.
9. Check θ_B from Equation 19.4 to see if cycle length chosen is satisfactory. If not, adjust and repeat Steps 1-6. (Note that "q" is independent of cycle length.)
10. Use the results of the above for regeneration calculations.

A similar approach may be used to check an existing tower or a bid proposed by a vendor.

Example 19.1: 0.27×10^6 std m^3 [10 MMscf/d] of a 0.6 relative density natural gas is to be dehydrated. The wet gas enters saturated at 6.9 MPa [1000 psia] and 38°C [100°F]. The vendor proposes a unit composed of 2-76 cm [30 in.] O.D. towers containing silica gel beds 4.57 m [15 ft] in length. After allowing for shell and internal insulation thickness, the bed diameter is 64.8 cm [25.5 in.]. Does this meet company criteria including a gas superficial velocity not exceeding 9.15 m/min [30 ft/min]? The water content of the inlet gas is 1021 kg 10^6 std m^3 [61 lb/MMscf]. $z = 0.88$ and bulk density of gel is 721 kg/ m^3 .

$$\text{Water absorbed} = (0.27)(1021)/3 = 91.9 \text{ kg/cycle}$$

$$\text{From Equation 19.9, } v_g = \frac{(307)(0.27)(0.88)(311)}{(6900)(0.648)^2} = 7.83 \text{ m/min}$$

$$\text{From Equation 19.3, } q = \frac{(0.054)(0.27)(1021)}{0.648^2} = 35.45 \text{ kg/h}\cdot\text{m}^2$$

$$\text{From Equation 19.2, } h_z = 141 \left(\frac{35.45^{0.7895}}{7.83^{0.5506} (100)^{0.2646}} \right) = 225 \text{ cm} = 2.25 \text{ m}$$

$$\text{From Figure 19.7, } x_s = (16)(0.9) = 14.4$$

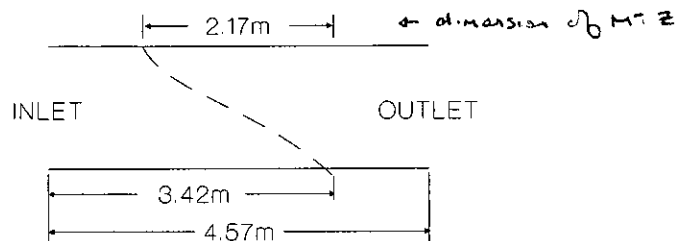
$$\text{From Equation 19.1, } x = \frac{14.4 [4.57 - (0.45)(2.25)]}{4.57} = 11.2 \text{ kg H}_2\text{O}/100 \text{ kg gel}$$

Example 19.1 (Cont'd.):

From Equation 19.5,
$$h_B = \frac{(127.4)(91.9)}{(721)(0.648)^2(11.3)} = 3.42 \text{ m}$$

From Equation 19.4,
$$\theta_B = \frac{(0.01)(11.3)(721)(4.57)}{35.45} = 10.5 \text{ hours}$$

This calculation shows the following situation:



Sometimes in this problem
we find that the total adsorbent
alters it in the calculation
M.T.E. etc.

When the desiccant is *new*, about 1.15 m of the bed will be below the front of the water zone when dehydration is discontinued about 2.5 hours before the zone front reaches the end of the bed. Thus, this unit will be able to easily produce pipeline specification gas initially, with a safety factor.

In the absence of abnormal contamination, how long can it do so? There are several approaches one can take. One can resolve Equation 19.4, for $\theta_B = 8$ hours, and find equivalent x . In the example, this $x = 8.6$. From Figure 19.4, the service life will be at least 24 months, and probably longer.

Some prefer to fix " x " based on experience and then use this value to fix h_B as total bed length from Equation 19.5, and then adjust cycle time from Equation 19.4 so that the two are compatible. Since any compatible combination of variables is satisfactory, various designs may be satisfactory.

I must emphasize that this is an approximate calculation. It is reliable enough though that if a proposed unit appears unsatisfactory when thus analyzed, further conversations with the vendor are indicated.

These equations are specifically for silica gel but may be used also for alumina and molecular sieves for dehydration of natural gas. The zone length, h_z , will be shorter, however, since both alumina or 4A and 5A sieve have a limited capacity for hydrocarbons. If hydrocarbons are already on the surface it lengthens the transfer zone length. As noted previously,

Alumina – h_z is 0.8 times that of silica gel.

4A or 5A sieve – h_z is 0.6 times that of silica gel.

All other variables may be estimated satisfactorily from the equations shown.

You may have noticed that the water loading calculation assumes all of the entering water vapor is removed from the bed. This is done because the exit water content is very low in a properly operating unit. It also varies somewhat with time. Thus, this assumption is convenient and introduces a small safety factor into the calculation.

REGENERATION AND COOLING CALCULATIONS

The problem is to supply enough carrier gas and heat to desorb the adsorbed components and then cool down the bed in the cycle time available. For a two-tower plant, both heating and cooling must be accomplished in the cycle time. With a three-tower system, with one tower drying at a time, twice the cycle length is available. For multiple towers (beyond two), the time depends on the switching pattern used.

The curve shapes of Figure 19.4 are typical of plant operations although the absolute and relative times shown vary with the installation. Curve 1 is the temperature of the gas to the bed being conditioned. Temperature " T_H " is the outlet gas temperature from the heater. Curve 2 is the outlet temperature of the regeneration gas from the bed. The difference in temperature between Curves 1 and 2 fixes the available heat energy available for transfer to the desiccant bed.

Temperature " T_4 " is the maximum regeneration temperature. It will vary from 175-300°C [347-572°F]. The lowest temperature possible is recommended. " T_H " should be at least 19°C [34°F] higher than " T_4 " and 38°C [68°F] is often desirable. In no case though is a value of " T_H " higher than 315°C recommended.

The total heat load is that necessary to heat the vessel and its contents to " T_4 " plus the *heat of desorption* of all adsorbed components. The adsorbed components are water, hydrocarbons and contaminants. The heat of desorption is the latent heat of vaporization plus that energy necessary to break the wetting forces. It is a function of concentration.

For gels the concentration of hydrocarbons ahead of the water front will be about 7-10 kg of hydrocarbon per 100 kg desiccant. Behind it, the concentration will fall to about 1-2 kg per 100 kg. Since the front will vary with water load and desiccant condition at the end of the adsorption process – which varies with time in an unknown way – it is normally satisfactory to assume that the hydrocarbon mass to be desorbed is about 10% of the water adsorbed, for cycles longer than four hours. For shorter cycles, a detailed hydrocarbon analysis is recommended. For sieves, hydrocarbon adsorption will be negligible.

The total regeneration cycle in Figure 19.4 is divided into four parts – A, B, C and D. Temperature " T_H " and regeneration gas flow rate govern the time for each. Total regeneration time cannot be greater than the time available which in turn is fixed by number of towers and θ_B for the drying tower(s). Almost all of the hydrocarbon will be removed in Interval A; almost all water in Interval B. From test, $T_2 \cong 110^\circ\text{C}$ [230°F], $T_3 \cong 127^\circ\text{C}$ [260°F] and $T_B \cong 116^\circ\text{C}$ [240°F]. Temperature " T_1 " is that of the entering wet gas.

The sensible heat load will include the desiccant, those adsorbed liquids not yet desorbed, the steel vessel, retaining screens, and support balls. A "guard bed" installed to protect gel from liquid water would be added to the desiccant weight.

Steel Shell

The heat required for the steel shell will depend on whether internal or external insulation is used. Internal insulation is of two types: (1) a steel "can" inside the shell that provides a stagnant gas space between the bed and shell or (2) cast or sprayed internal insulation. With internal insulation the bed diameter usually is about 15 cm [6 in.] less than the shell I.D..

Internal insulation is a requirement on towers operating on cycles less than several hours. It may be desirable on longer cycle units to save on fuel costs. The shell sensible heat load is about 0.50-0.75 that of an externally insulated tower.

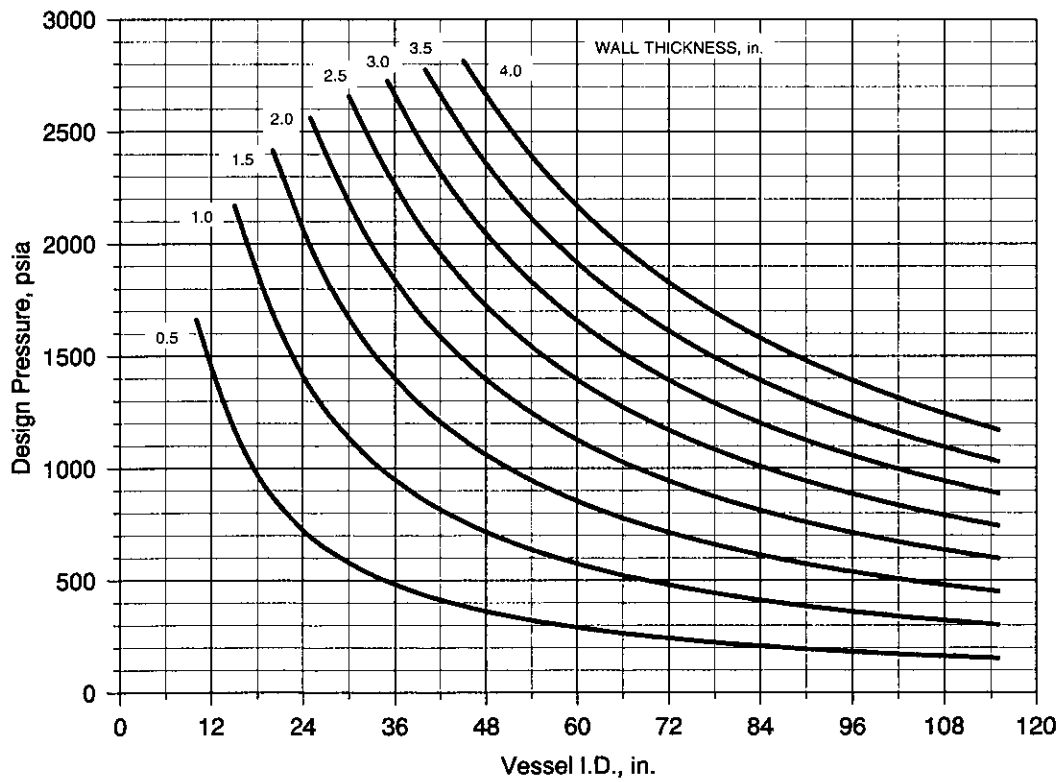
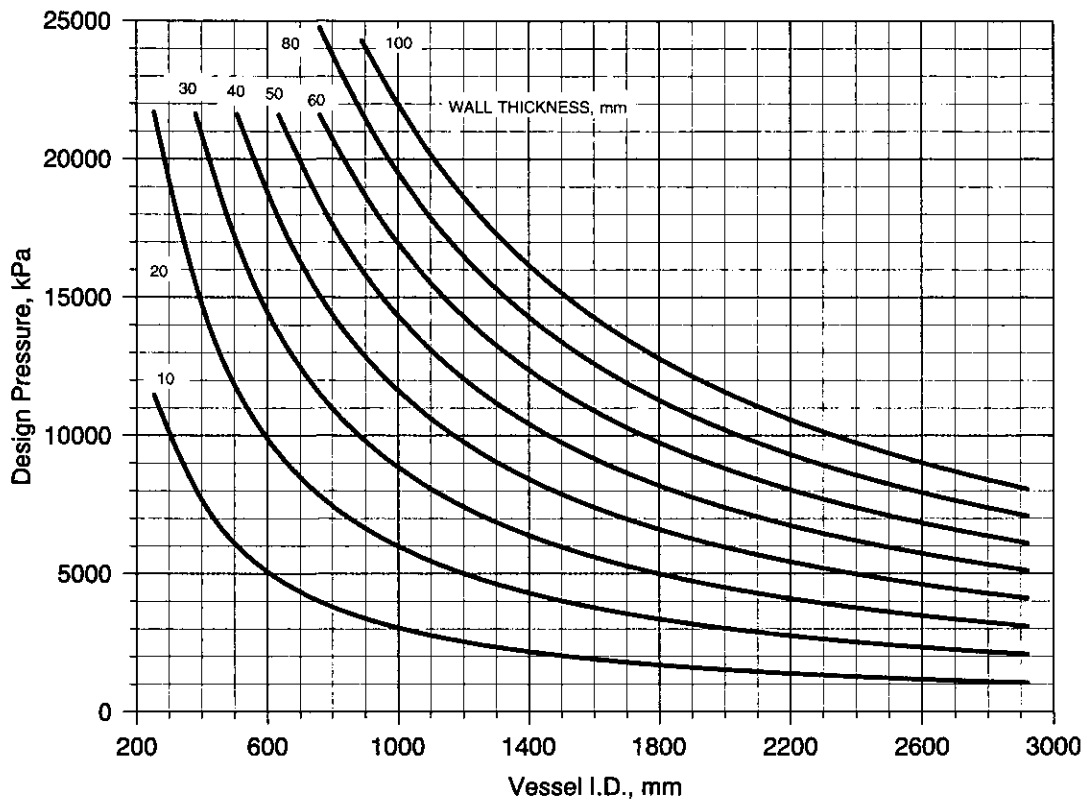


Figure 19.8 Approximate Pressure Vessel Wall Thickness

The approximate thickness of shell is shown in Figure 19.8. The diameter should be the nearest commercial size available that will allow a desiccant bed diameter at least as large as that calculated. The mass of the shell and heads may be estimated by Equation 19.13.

$$m = A h d t \quad (19.13)$$

Where: m = mass
 h = vessel length (HS - HS)
 d = vessel I.D.
 t = shell thickness
 A = weight factor
 HS - HS = head seam to head seam

Metric	English
kg	lbm
m	ft
mm	in.
mm	in.
0.0347	15

Heat Factors

The following values are suitable for the procedure below.

Heat Capacity, C_p

Steel - 0.50 kJ/(kg·K) [0.12 Btu/(lbm·°F)]

Liquid Water = 4.19 kJ/(kg·K) [1.0 Btu/(lbm·°F)]

Desiccant - from table on page 368

The heat of desorption is larger than the latent heat because of the forces holding the adsorbate to the adsorbent.

Heat of Desorption

Water on molecular sieves - 4187 kJ/kg [1800 Btu/lbm]

Water on alumina and gel - 3256 kJ/kg [1400 Btu/lbm]

Adsorbed hydrocarbon - 465 kJ/kg [200 Btu/lbm]

Heating from T_1 and T_2 (Interval A) -

The total heat load is the sum of the following:

1. (Wt. desiccant)(C_p)($T_2 - T_1$)
2. (Wt. hydrocarbon)(C_p)($T_A - T_1$)
3. (Wt. vessel shell)(C_p)($T_2 - T_1$)*
4. (Wt. inert balls)(C_p)($T_2 - T_1$)
5. (Wt. hydrocarbons)(heat of desorption)
6. (Wt. water)(C_p)($T_2 - T_1$)

*With internal insulation multiply shell weight by 0.70.

If "m" is the regeneration gas mass flow rate, the available energy to supply the above needs is:

$$(m)(C_p)(T_H - T_A)(\theta_A) \quad \text{or} \quad (m)(\Delta h)(\theta_A) \quad (19.14)$$

Heating from T_2 to T_3 (Interval B) -

Total heat is found as follows:

1. (Wt. desiccant)(C_p)($T_3 - T_2$)
2. (Wt. water)(C_p)($T_B - T_2$)
3. (Wt. water)(Ht. of adsorption)
4. (Wt. inert balls)(C_p)($T_3 - T_2$)
5. Vessel load - same procedure as Interval A, with new temperatures.

The total available energy is found from the equation

$$(m)(C_p)(T_H - T_B)(\theta_B) \quad \text{or} \quad (m)(\Delta h)(\theta_B) \quad (19.15)$$

Heating from T_3 and T_4 (Interval C) -

This step merely is for the purpose of removing the "heel" from the bed -- a final clean-up of heavier components. The total heat load is then

1. (Wt. desiccant)(C_p)($T_4 - T_3$)
2. (Wt. inert balls)(C_p)($T_4 - T_3$)
3. Vessel heat load consistent with previous intervals.

Available heating

$$(m)(C_p)(T_H - T_C)(\theta_C) \quad \text{or} \quad (m)(\Delta h)(\theta_C) \quad (19.16)$$

Cooling from T_4 and T_5 (Interval D) -

Temperature " T_5 " is normally not less than 50-55°C [122-131°F] to minimize presaturation of the desiccant. The coolant is presumed to be gas at temperature " T_1 ."

1. (Wt. of tower and contents)(C_p)($T_4 - T_5$)

The cooling available is

$$(m)(C_p)(T_D - T_1)(\theta_D) \quad \text{or} \quad (m)(\Delta h)(\theta_D) \quad (19.17)$$

assuming cooling gas rate is equal to the regeneration gas rate.

For each interval, as well as the entire cycle, energy available for heating and cooling must equal the load. One has four equations containing "m" and " θ ," the total value of each equation (from the corresponding heat load) and the fact that $\theta_A + \theta_B + \theta_C + \theta_D$ must not exceed the time available. For a given cycle time, a solution for "m" is possible.

Normally, the maximum heat load occurs in Interval B. As a first try it might be assumed that "m" is about 10% of the main gas flow rate.

During the desorption of water (Interval B) the combination of gas flow rate and its water capacity must be such that the gas can hold the water desorbed in time θ . If it is assumed that the water comes off at an average temperature of 116°C [241°F], the saturated water content of the gas at this temperature and regeneration pressure fixes the total amount of water vapor the gas can hold.

From a material balance

$$\text{Mass of water adsorbed/cycle} = \frac{18 m}{\text{MW Gas}} (\theta) (y_B - y_1) \quad (19.18)$$

Where: θ = time for water desorption, hours

y_B = mole fr. of water in saturated gas at T_B

y_1 = mole fr. of water in entering wet gas (or dry gas when used for regeneration)

The time for water desorption is about 16-17% of the total cycle length for cycles of 4 hours or longer. This may be used to estimate a value of "m" from Interval B, the most critical interval of the cycle.

The regeneration gas rate fixed by Equation 19.18 is a minimum rate and will normally be less than "m" required to meet total regeneration heat needs.

Heater Load

$$Q = (m)(C_p)(T_H - T_1) = m \Delta h \quad (19.19)$$

Where: m = regeneration gas rate
 C_p = gas heat capacity
 T_H = heater outlet temperature
 T_1 = heater inlet temperature
 Δh = gas enthalpy change at unit from T_1 to T_H

Do not skimp on heater size! A unit at least 25% higher than that calculated should be used. In this service a heater thermal efficiency of 70% is appropriate to find fuel consumption.

Condenser Load

One needs to calculate the condenser load for all three intervals to find the highest load. It will normally occur in interval B for long cycle units. The latent heats of water and hydrocarbons may be estimated in Chapter 8. Knowing the time for the interval and assuming the desorption is uniform during it, one can find the latent heat load. To this one must add the gas sensible heat load. The normal temperature approach will be 16-20°C [29-38°F] for air cooling and 8-10°C [15-18°F] for water cooling.

Summary

The above heat balance must be applied for every cycle length and configuration used in adsorber tower sizing. From this one can develop an array of information useful for planning and decision purposes.

The actual unit will possess a diameter and length different than that calculated to conform to commercially available steel sizes and fabrication economics.

HYDROCARBON RECOVERY

Short cycle unit
 The basic mechanism for hydrocarbon recovery is similar but more complex than dehydration. One is faced with describing multiple zone behavior.

In a dehydrator, the purpose of the condenser is just to remove the desorbed liquids from the gas stream. In the short cycle plant, condenser operation has a critical effect on recovery. The adsorption bed simply serves to concentrate the recoverable components so that condensation is more efficient. The temperature and pressure of condensation is a critical parameter governing plant economics. A carbon plant with a refrigerated condenser is capable of good ethane recovery in some instances. A gel plant with ambient condensation is limited to some butane recovery and 75-90% of the pentanes. There are many alternatives in between.

Consider that the hydrocarbon deposited on the bed is picked by 10-15% as much regeneration gas. The net effect is to make the gas 6-10 times richer in condensable hydrocarbons, thus making recovery easier. A refrigerated condenser may be used to further enhance recovery but the condensation temperature should exceed the hydrate temperature. In some cases enough liquid may be recovered to reduce simultaneous dehydration cost below that of glycol.

Ambient cooling plants have been used primarily on lean gas streams where other methods of processing were not economically attractive. The untapped potential for hydrocarbon dewpoint control and as an adjunct to refrigeration appears large particularly at higher pressures. The biggest obstacle to more widespread use of the process would appear to be the rather unimaginative design methods used to date.

Process Characteristics

The capacity of most desiccants is about the same for hydrocarbons as for water. Activated carbon, of course, has no effective capacity for water.

Figure 19.9 shows the equilibrium capacity of silica gel for various hydrocarbons in a two component gas where the second component is methane. The figure shows both static equilibrium (from cell tests) and dynamic equilibrium (from flow tests). It is apparent that capacity is not affected very much by flow.

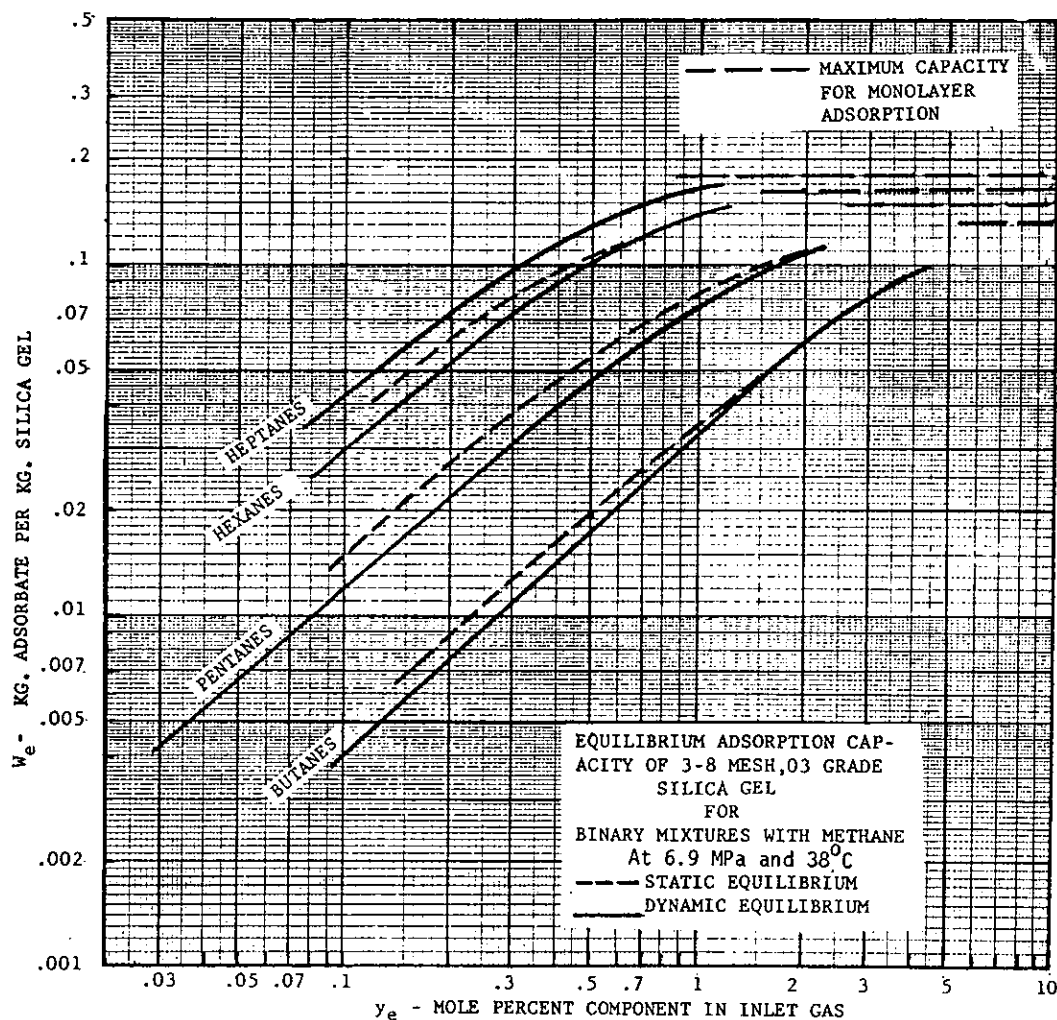


Figure 19.9 Hydrocarbon Equilibrium on Silica Gel

Notice that all curves are approaching the monolayer capacity of the gel. The monolayer capacity is found by assuming that only one layer of molecules is held to the solid surface. Knowing both surface area and molecule size, one may compute the capacity. This same characteristic has been found for all gels, aluminas and molecular sieves. This means that ultimate capacity for any component is fixed by surface area — provided that the component is small enough to enter the interior of the adsorbing particle.

Actual capacity for any component is fixed by the zone movement previously described, bed geometry, equilibrium capacity and gas flow rate. The surface of the adsorbent is always occupied by some molecule. As the zone of a given component progresses down the bed it must displace the molecules already there. The rate of displacement depends on their relative wettability.

Theoretically, the zone for any component cannot move any faster than it can completely displace the materials ahead of it. In actual practice, at commercial flow rates, the zone tends to "over-run" said displacement. Therefore, true chromatographic separation does not occur. This is illustrated in Figure 19.10.

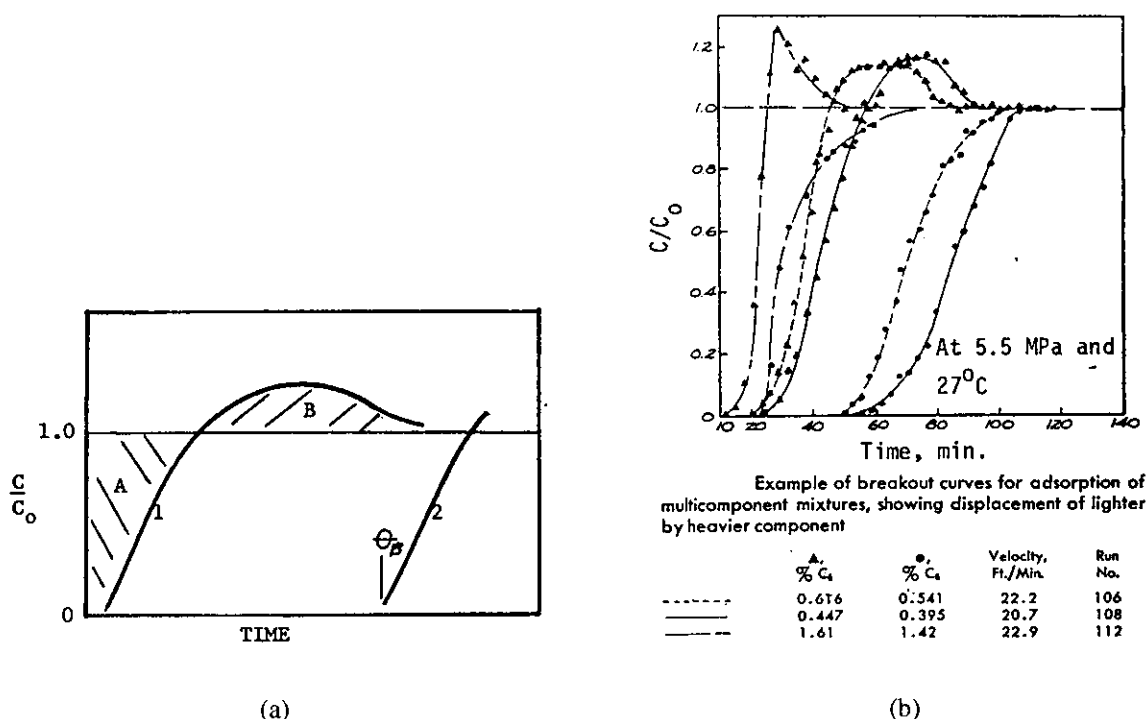


Figure 19.10 Illustration of the Adsorption/Desorption Process

As shown in Part (a), once the front of a given zone reaches the outlet of the bed the ratio of outlet to inlet concentration (C/C_0) starts to increase. When this ratio reaches unity, all primary adsorption ceases for that component. Desorption now begins because the zone behind it is replacing the material adsorbed. The concentration ratio rises above unity but again *approaches* unity when this next zone starts breaking out the end of the bed. This process continues until the cycle is terminated.

Area A is representative of the amount of Component 1 adsorbed and Area B of that amount desorbed by the next zone. The latter is smaller than the former. At times θ_B for Component 2 (Area A - Area B)/Area A is generally about 0.35-0.40. Thus, immediate displacement has not occurred.

This is shown by the following test on a field unit.

Inlet Mol% (C_0)	Time, min.	Outlet mol %, C		
		C_3	iC_5	nC_5
—	—	1.020	0.120	0.085
2	2	0.804	0.031	0.012
12	12	0.976	0.074	0.031
22	22	0.938	0.075	0.043
32	32	0.962	0.077	0.065
42	42	0.970	0.155	0.130
52	52	0.952	0.109	0.102

Even if one recognizes the 6-10% error in sampling and analysis, no sharp separation has occurred. The propane has broken through in less than two minutes. This is not a very efficient plant because even

good pentanes recovery is not obtained early in the cycle. Beyond this point, the exit stream is being *enriched*.

If liquid recovery is the goal, some net amount of component is available even after its zone passes from the tower. The recovery will simply be less. If hydrocarbon dewpoint control is the goal, such enrichment is probably intolerable for components heavier than the butanes. In such case, the iso-pentane θ_B represents maximum cycle time. For these reasons, liquid recovery plants tend to use longer cycles than dewpoint control plants since the loss of efficiency is not so apparent. Unfortunately, the bulk of the recovery plants I have examined use cycle times too long for most efficient recovery. Recovery is not limited by the process but by the way it is applied.

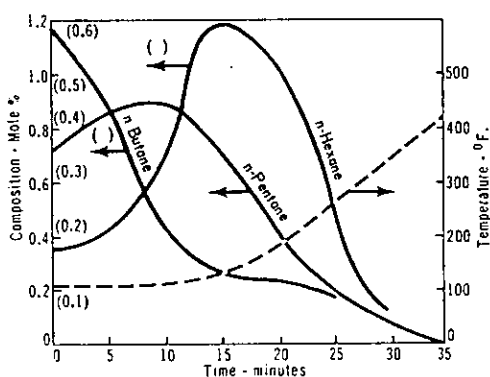
Part (b) of Figure 19.10 shows the adsorption-desorption process for a test run. The gas in question contained methane, ethane, negligible propane and butanes, and the amounts of pentanes and hexanes shown. There was no heavier component present to displace the hexanes. Water content was negligible.

With activated carbon the zones tend to move slower. For one thing, water does not promote displacement. Basically though, the zone speed is lower because of the greater affinity for the lighter hydrocarbons.

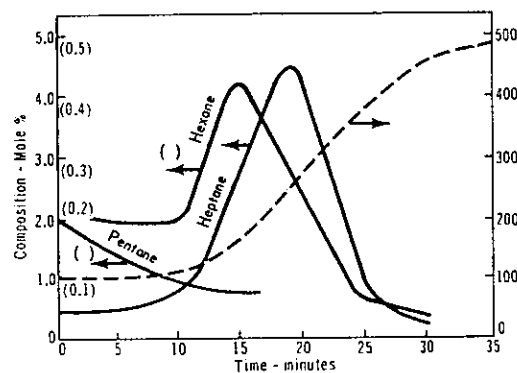
Regeneration and Recovery

Note variation in composition of regeneration gas

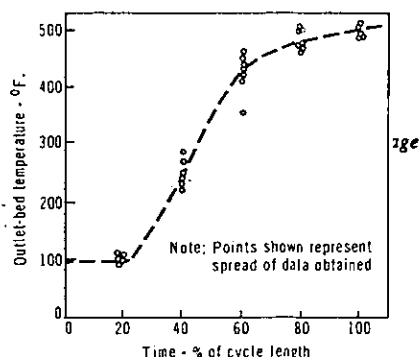
Figure 19.11 summarizes the regeneration behavior of a short cycle plant. Notice that the materials do not desorb at a constant rate. The pentanes and lighter start desorbing almost immediately. The hexanes and heavier concentration in the exit gas peak after a finite time.



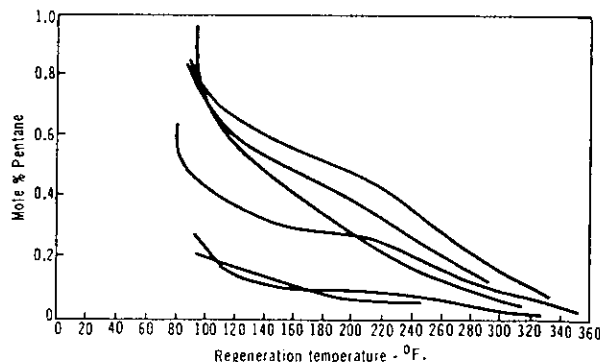
With butane, pentane, hexane



With pentane, hexane, heptane



Outlet bed temperature produces characteristic curve



Characteristic pentane content of regeneration gas

Figure 19.11 Typical Regeneration for Hydrocarbon Adsorbers

Obviously, the composition of the stream to the condenser varies continually with time. For this reason a series of flash calculations must be made with time to accurately represent the liquid recovery to be expected. The minimum necessary amount of regeneration gas should be used to enhance the concentration of recoverable components in the condenser. The use of refrigeration will likewise enhance recovery. Even at refrigeration levels as high as 16-20°C [60-68°F], marked recovery efficiency may be obtained.

General Specification

Proper detailed design is complicated enough that a computer is necessary. A design method will not be outlined herein although the principles governing it have been. It is sufficient to say that the general empirical methods to date fail to utilize the full potential of the adsorption process.

Most obvious problems stem from improper specification. A liquid recovery guarantee is useful for planning purposes but is more likely to be a basis for controversy than a measure of plant performance.

I recommend the following minimum specifications:

1. Gas flow limited by the rate previously discussed for dehydration (to promote desiccant life).
2. Cycle length should be fixed by the following considerations:
 - a. Not less than 15 minutes for gas containing pentanes and heavier.
 - b. Breakthrough time for the component for which recovery is desired or which must be removed for dewpoint control.
3. Bed length should be at least 15 feet.
4. Regeneration should be to at least 230°C [446°F] and preferably 260°C [500°F] when processing gases containing pentanes and heavier.
5. The alternative of using refrigeration instead of ambient cooling in the condenser should be considered.

Items 1-4 are not independent; each affects the other. If the cycle length is less than 15 minutes it is almost impossible to remove the "heel" properly, which ultimately affects the economics adversely. If this 15 minutes is greater than the breakthrough time for key component, some compromise is needed. Breakthrough time depends on gas velocity and bed length (for a given gas composition and adsorbent. Economics and/or process needs will govern the compromise. As a matter of information – so one may make an intelligent decision – it is wise to also specify that the vendor furnish you with adsorption efficiency as well as condenser recovery. Adsorption efficiency is simply that fraction of the component entering during the proposed cycle length that is retained on the adsorbent. This enables you to not only compare the relative merit of competitive bids but to make necessary changes prior to purchase. Any design approach that uses an overall method of calculation from inlet gas to stock tank may yield a workable plant but seldom an optimum one.

The adsorbent bed may contain more than one adsorbent. Such composite bed will behave like two towers in series and should be treated as such.

LIQUID DEHYDRATION

The gels, alumina and molecular sieves may be used to dry hydrocarbon liquids. The flow sheet is similar to that for gas. Some larger plants are designed so that the flow may be reversed to "loosen" the bed if it has been compacted or to free the retaining screens of sediment. This provision is seldom needed for fractionated liquids. If there is any possibility of free liquid water being present, a free water knockout should be provided.

The primary difference is in the regeneration cycle. Several systems are commonly used to provide regeneration:

1. Gas
2. Steam (closed)
3. Closed Vapor

Table 19.3 summarizes these processes. The gas system may use superheated steam, natural gas, flue gas or any inert gas. If steam is indicated, the closed system is normally recommended. Even super-heated steam tends to rehydrate the desiccant. With alumina a monohydrate is formed which is less efficient than the trihydrate.

TABLE 19.3
Comparison of Various Regeneration Practices

Method	Advantage	Disadvantages	Common Usage
Open Steam	<ol style="list-style-type: none"> 1. Low initial cost 2. Simple in design 	<ol style="list-style-type: none"> 1. Usually shows highest operating cost. 2. Only sensible heat may be utilized. 3. Tendency to rehydrate desiccant beds. 4. Requires cheap source of steam. 	General service where steam is available at low cost. In conjunction with refineries and gasoline plants for relative nonvolatile liquids.
Closed Steam	<ol style="list-style-type: none"> 1. Low operating cost. 2. Either saturated or superheated steam may be used. 3. Little safety hazard. 4. No contamination of product. 	<ol style="list-style-type: none"> 1. High initial cost. 2. Requires cheap source of steam. 	Same as above.
Natural Gas, Flue Gas, etc.	<ol style="list-style-type: none"> 1. Uses readily available material. 2. Low operating cost. 3. Simple construction. 4. Readily adaptable to automatic control. 	<ol style="list-style-type: none"> 1. Has higher operating cost than closed steam system. 2. Introduces some additional safety hazard. 3. Requires compressor if high pressure gas not available. 	Field locations, Product Pipe lines, Non-Volatile Liquids.
Closed Vapor	<ol style="list-style-type: none"> 1. Simple operation. 2. Low cost of operation. 3. Minimizes loss of expensive volatile liquids. 4. No contamination of product. 	<ol style="list-style-type: none"> 1. Control of system more critical. 2. Requires pumping equipment. 3. Requires efficient condensation of exit regeneration gas. 	With volatile liquids such as propane, butane, etc. Where composition of feed is substantially constant.

The gas is heated to 160°C [320°F] in a heater in the normal manner.

A closed stream system is recommended over the open stream system when steam is to be used. The initial cost of imbedding steam coils is greater but so is the efficiency. The latent heat of saturated steam may be utilized. Contamination of the *hold-up stock* in the drying tower at the end of the cycle is also minimized. When drying volatile material such as propane and butane, they may be vaporized for use as regeneration gas. The hold-up stock is normally used for this purpose.

Design Considerations

The solubility of water in sweet hydrocarbons is shown in Figure 19.12. Notice that it is far more soluble in many unsaturated and aromatic hydrocarbons than in the normal paraffins. Knowledge of composition is thus very important. The presence of sulfur compounds enhances water solubility. If no data are available for the specific liquid, a *weight fraction* relationship may be used to estimate liquid mixture water content.

Most contracts specify that the dried liquid show a negative result to the Cobalt Bromide test, which is equivalent to a water content of 15-30 ppm. This is virtually bone dry for ppm is weight percent times 10 000. In design, one assumes all incoming water is removed.

Liquid velocity should be 1-2 m/min. This will fix tower diameter. Tower length will usually be shorter than for gas. An (L/D) ratio of 2-3:1 is common. As little as three seconds contact time is commonly provided. Some operators require a minimum bed length of 1.5 meters.

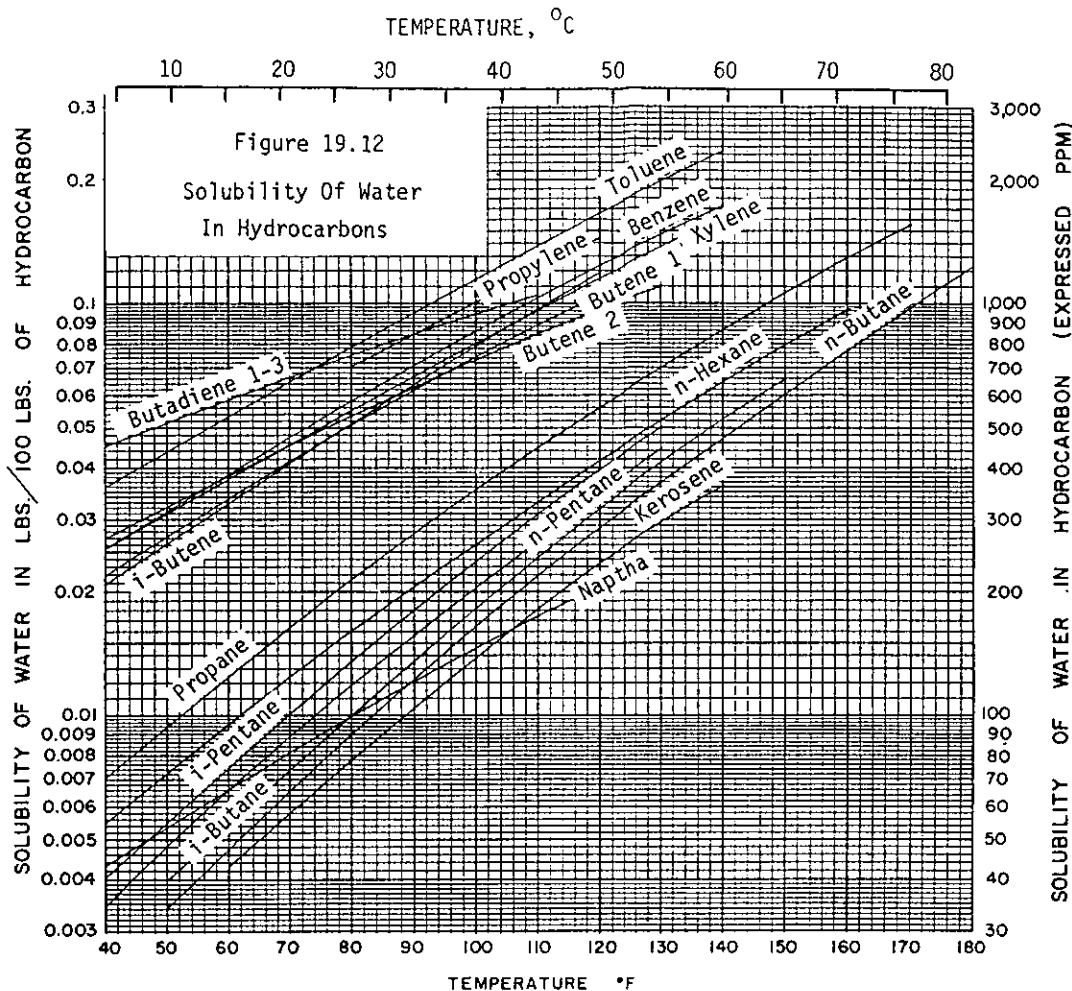


Figure 19.12 Solubility of Water In Hydrocarbons

Activated alumina has been used quite widely for liquid drying since it is relatively inexpensive and tower costs are minimized at low pressure. An effective capacity of 4-5 kg of water per 100 kg alumina is common. This is equivalent to that in gas service, as is the capacity of gels and sieves.

A 3A sieve is useful for drying liquids where contaminants are present, for the opening size is too small to admit these contaminants to the interior particle surface. Olefins, for example, may "tie-up" the available surface and reduce water capacity when using alumina.

Molecular sieves may be used also to both dry and sweeten liquids as discussed in a later section.

GAS AND LIQUID SWEETENING

Molecular sieves are widely used for sweetening gases and liquids. Carbon dioxide and sulfur compounds are co-adsorbed with water.

Figure 19.13 is a basic, simple flow sheet for a sweetening unit. The process is much like that for water alone except for handling the regeneration gas. Burning the flare gas as shown may be in conflict with environmental standards. As an alternative, the regeneration gas may be treated first by an amine type

sweetening process after which it is dehydrated using glycol. The effluent gas from the dehydrator is then co-mingled with the effluent gas from the molecular sieve beds. Thus, no gas is flared.

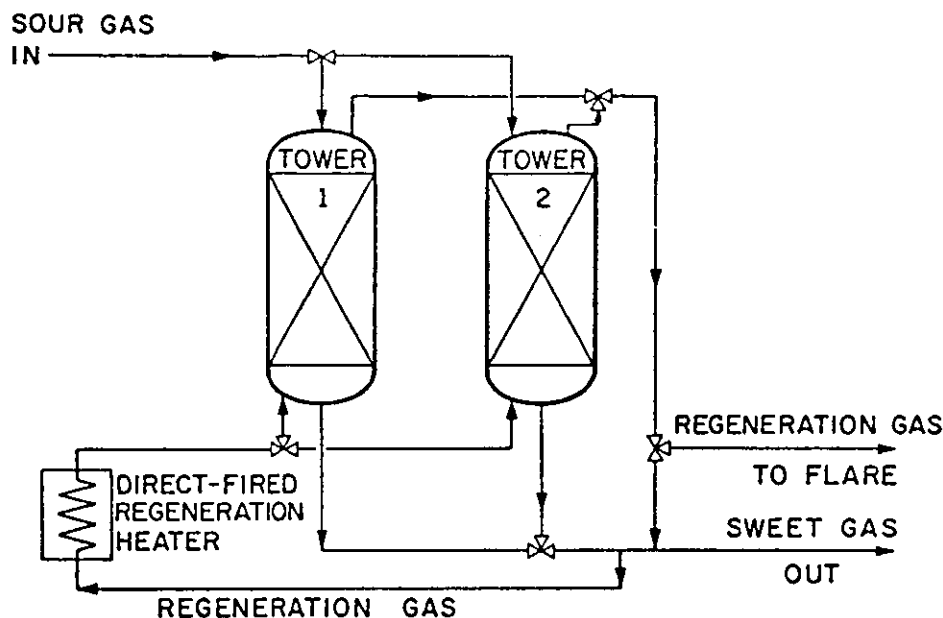


Figure 19.13 Flow Diagram for Basic Molecular Sieve Drying and Sweetening Process

A closed cycle process known as the EFCO system is shown in Figure 19.14. It is also designed to reduce the problem of flaring sour gas.

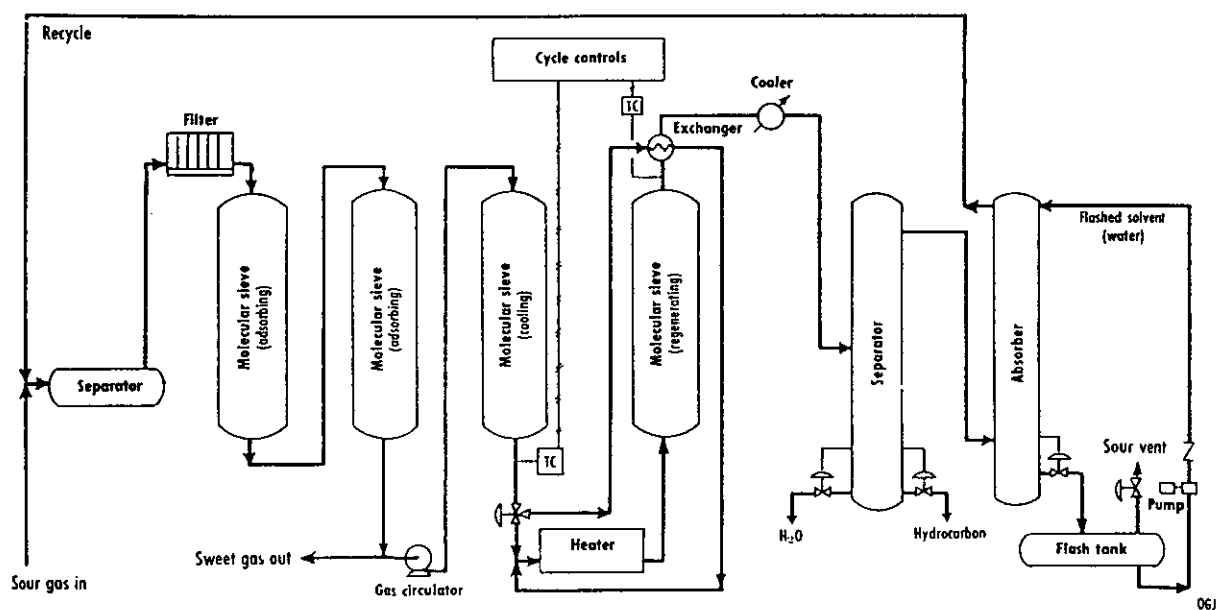


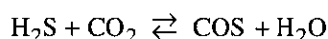
Figure 19.14 Flow Diagram for the EFCO Process

With sour gases the major use of molecular sieves is to selectively remove H_2S and other sulfur compounds from gases containing large amounts of CO_2 . Most processes like the amines are non-selective. One must remove all of the CO_2 to meet sulfur specifications. This is not economical. Only that amount of CO_2 should be removed to meet heating value or Wobbe number specifications. Sieves have an affinity for CO_2 but it is greater for polar molecules like H_2O and H_2S . The cycle can be adjusted so that the amount of CO_2 removed, if any, can be controlled.

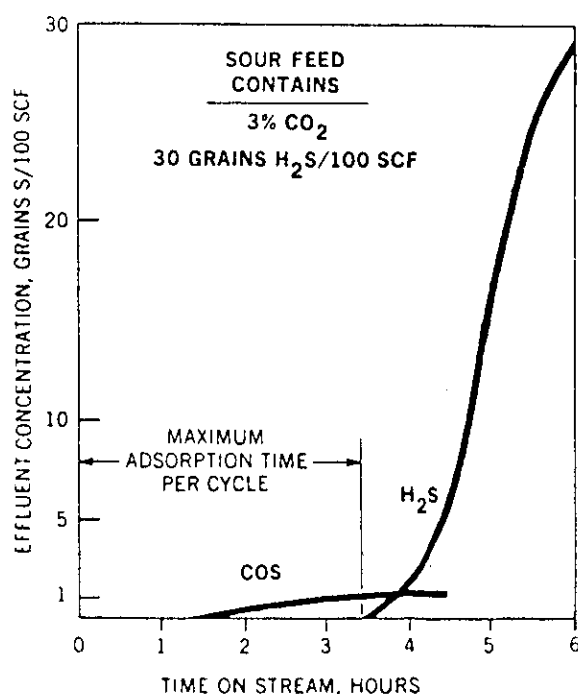
Sieves also can be used for removal of mercaptans and other heavier sulfur compounds. In prepurification of gases for LNG plants, water, H_2S and CO_2 can be reduced to acceptable levels.

Use of 13X sieve for the simultaneous drying and sweetening of propane and butane has been increasing. This replaces drying first, and then sweetening, by a process like caustic wash.

One problem that can occur is the conversion of H_2S to carbonyl sulfide (COS) in the bed. This is the result of the equation



The equilibrium constant for the reaction to the right increases with temperature. The figure following shows how COS forms with time. This is important for on occasions the effluent has been sweet in H_2S but contained too much COS.



*Approx. 1% COS
 1.2% COS since
 adsorption on
 of H₂S
 propane
 ... with
 H₂S
 COS generated
 propane +
 ready water
 water*

In the presence of water the COS is hydrolyzed back to H_2S . Obviously, water vapor thus tends to keep COS from forming.

Further details about all sweetening processes are shown in Volume 4, "Gas and Liquid Sweetening," a companion book in the Campbell Petroleum Series.

Rather obviously, molecular sieves are the most versatile of the commercial desiccants now available. They also are much more expensive. Proper choice of desiccant for maximum economic benefit is not routine. Arbitrarily using molecular sieves, regardless of cost, is not the automatic answer. One must look at all factors. In addition to the initial cost, molecular sieves also have a higher heat of desorption which thus increases heat loads for regeneration.

Either alumina or gel may be suitable in a given case for either gas or liquid drying alone. For liquid drying, alumina may be the most economical if the liquid is already sweet. The lower capacity is offset by the lower water loading, which results in reasonable size towers.

Where suitable, gel is a commonly used material for the drying of sweet gases. It may be used for sour gases but the co-adsorption of heavy or aromatic hydrocarbons may lead to regeneration problems. For the drying of sweet gas or recovery of the pentanes plus, gel may prove to be a suitable economic compromise between cost and performance. In drying sour gases where the pH of the adsorbed water is less than 5.0, the proper choice is acid resistant molecular sieves.

Somewhat by default, molecular sieves will probably be the choice for

1. Drying fluids at a temperature above about 50°C.
2. Drying liquids when heavy hydrocarbons and/or aromatics are present which can reduce the capacity of alumina gels for ethane, propane and butane.
3. Co-adsorption of water and sulfur compounds.
4. Selective adsorption of H₂S from high CO₂ gases.
5. Acid gases when the pH of the adsorbed water is less than 5.0.
6. Outlet dewpoints less than -73°C.

By careful selection of the desiccant and optimum design of the system, dry desiccant can prove to be economically attractive when compared to the use of glycol or hydrate inhibition.

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NOTES:

APPENDIX A

GENERAL CONVERSION FACTORS

Base SI metric units

Quantity	Name	Symbol
Length	meter	m
Mass	kilogram	kg
Time	second	s
Electric current	ampere	A
Thermodynamic temperature	kelvin	K
Luminous intensity	candela	cd
Amount of substance	mole	mol

Important derived and supplementary SI units

Quantity	Name	Symbol	Formula
Electric capacitance	farad	F	A-s/V
Electric charge	coulomb	C	A-s
Electric conductance	siemens	S	A/V
Electric inductance	henry	H	Wb/A
Electric potential	volt	V	W/A
Electric quantity	coulomb	C	A-s
Electric resistance	ohm	Ω	V/A
Electromotive force	volt	V	W/A
Energy	joule	J	N-m
Energy flux	watt	W	J/s
Force	newton	N	kg-m/s ²
Frequency	hertz	Hz	1/s
Illuminance	lux	lx	1m/m ²
Luminous flux	lumen	lm	cd-sr
Magnetic flux	weber	Wb	V-s
Magnetic flux density	tesla	T	Wb/m ²
Power	watt	W	J/s
Pressure	pascal	Pa	N/m ²
Quantity of heat	joule	J	N-m
Solid Angle	steradian	sr	---
Work	joule	J	N-m

SI prefixes and multiplication factors

Prefix	SI Symbol	Factor
tera	T	10 ¹²
giga	G	10 ⁹
mega	M	10 ⁶
kilo	k	10 ³
hecto	h	10 ²
deka	da	10 ¹
deci	d	10 ⁻¹
centi	c	10 ⁻²
milli	m	10 ⁻³
micro	μ	10 ⁻⁶
nano	n	10 ⁻⁹
pico	p	10 ⁻¹²
femto	f	10 ⁻¹⁵
atto	a	10 ⁻¹⁸

To convert from	to	Multiply by
acre	hectare	0.40469
acre	meter ²	4,046.9
acre-feet	meter ³	1,233.5
ampere-hour	coulomb	3,600.0
angstrom	nanometer	0.10000
astronomical unit	gigameter	149,598
atmosphere	bar	1.01325
atmosphere (normal)	kilopascal	101,325
atmosphere (normal)	pascal	101,325
atmosphere (1 kg/cm ²)	pascal	98,067
bar	kilopascal	100,000
bar	pascal	100,000
barrel (42 US gal)	meter ³	0.15899
barrel/acre-foot	decimeter ³ /meter ³	0.12889
barrel/day	decimeter ³ /second	0.00184
barrel/day	meter ³ /day	0.15899
barrel/day (1 kg/dm ³)	tonne/annum	58.0304
barrel/foot	meter ³ /meter	0.52161
barrel/hour	decimeter ³ /second	0.04416
barrel/hour	meter ³ /hour	0.15899
barrel/US ton	meter ³ /tonne	0.17525
board-foot	meter ³	0.00236
British thermal unit	joule	1,055.1
British thermal unit	kilojoule	1.05506
British thermal unit	watt-hour	0.29288
BTU/bhp-hour	watt/kilowatt	0.39301
BTU/foot ²	kilojoule/meter ²	37.2590
BTU/foot ²	kilowatt-hour/meter ²	0.01035
BTU/US gallon	kilojoule/meter ³	278.716
BTU/US gallon	kilowatt-hour/meter ³	0.07742
BTU/hour	watt	0.29307
BTU/minute	watt	17.5843
BTU/second	kilowatt	1.05506
BTU/hour-foot ²	watt/meter ²	3.15459
BTU/hour-foot ²	watt/meter ²	10.3497
BTU/hr-foot ² -°F	watt/meter ² -kelvin	5.67826
BTU/second-foot ² -°F	kilowatt/meter ² -kelvin	20.4418
BTU/hour-foot ² -°F	watt/meter ² -kelvin	18.6295
BTU/second-foot ² -°F	kilowatt/meter ² -kelvin	67.0661
BTU/hour-foot ² -°F/foot	watt/meter-kelvin	1.73074
BTU/hour-foot ² -°F/inch	watt/meter-kelvin	0.14428
BTU/second-foot ² -°F/in.	watt/meter-kelvin	519.220
BTU/pound mass	joules/gram	2.32600
BTU/pound mass	watt-hour/kilogram	0.64611
BTU/pound mol	joules/mol	2.32600
BTU/pound-°F	joules/gram-kelvin	4.18680
BTU/pound mol-°F	joules/mol-kelvin	4.18680
bushel	decimeter ³	35.2391
bushel	meter ³	0.03524
calorie	joule	4.18400
calorie/pound	joule/kilogram	9.22414
centipoise	newton-second/meter ²	0.00100
centipoise	pascal-second	0.00100
centistroke	millimeter ² /second	1.00000
chain	meter	20.1168
cycle/second	hertz	1.00000
darcy	micrometer ²	0.98692
degree (angle)	radian	0.01745
°F/100 ft	millikelvin/meter	18.2269
°F-foot ² -hour/BTU	kelvin-meter ² /watt	0.17611
dyne	millinewton	0.01000
erg	microjoule	0.10000
erg	millijoule	0.00010
fathom	meter	1.82880
foot	centimeter	30.4800
foot	meter	0.30480
foot/day	meter/day	0.30480
foot/°F	meter/kelvin	0.54864
foot/hour	meter/hour	0.30480
foot/hour	millimeter/second	0.08467
foot/minute	centimeter/second	0.50800
foot/minute	meter/minute	0.30480
foot/minute	meter/second	0.00508
foot/second	meter/second	0.30480
foot/second ²	meter/second ²	0.30480
foot ²	centimeter ²	929.030
foot ²	meter ²	0.09290
foot ² /hour	millimeter ² /second	25.8064
foot ² /second	millimeter ² /second	92.903
foot ³	decimeter ³	28.3169
foot ³	meter	0.02832
foot ³ /day	decimeter ³ /second	0.00033
foot ³ /day	meter ³ /day	0.02832
foot ³ /day	meter ³ /hour	0.00118

GENERAL CONVERSION FACTORS

To convert from	to	Multiply by
foot ³ /foot	meter ³ /meter	0.09290
foot ³ /hour	decimeter ³ /second	0.00787
foot ³ /hour	meter ³ /hour	0.02832
foot ³ /minute	decimeter ³ /second	0.47195
foot ³ /minute	meter ³ /minute	0.02832
foot ³ /pound	meter ³ /kilogram	0.06243
foot ³ /pound mol	meter ³ /kilomol	0.06243
foot ³ /second	meter ³ /second	0.02832
footcandle	lux	10.0764
footcandle-second	lux-second	10.0764
foot-pound force	joule	1.35582
foot-pound/US gallon	kilojoule/meter ³	0.35817
foot-pound/minute	milliwatt	22.5970
foot-pound/minute	watt	0.02260
foot-pound/second	watt	1.35582
gallon US liquid	meter ³	0.00379
gallon US/foot	meter ³ /meter	0.01242
gallon US/horsepower-hr.	decimeter ³ /megajoule	1.41009
gallon US/horsepower-hr.	millimeter ³ /joule	1.41009
gallon US/1,000 barrel	centimeter ³ /meter ³	23.8095
gallon US/foot	decimeter ³ /meter ³	133.681
gallon US/hour	decimeter ³ /second	0.00105
gallon US/mile	decimeter ³ /100 km	235.215
gallon US/minute	decimeter ³ /second	0.06309
gallon US/minute	meter ³ /hour	0.22712
gallon US/ton US	decimeter ³ /tonne	4.17270
grain	milligram	64.7989
grain/gallon US	gram/meter ³	17.1181
grain/100 foot ³	milligram/meter ³	22.8835
horsepower	kilowatt	0.74570
horsepower (boiler)	kilowatt	9.80950
horsepower/foot ²	kilowatt/meter ³	26.3341
horsepower-hour	kilowatt-hour	0.74570
horsepower-hour	megajoule	2.68452
inch	centimeter	2.54000
inch	millimeter	25.4000
inch	pica	6.02250
inch	point	72.2700
inch Hg (60°F)	kilopascal	3.37685
inch H ₂ O (60°F)	kilopascal	0.24884
inch/minute	centimeter/minute	2.54000
inch/second	centimeter/second	2.54000
inch/second	millimeter/second	25.4000
inch/year	millimeter/annum	25.4000
inch ²	centimeter ²	6.45160
inch ²	millimeter ²	645.160
inch ² /second	millimeter ² /second	645.160
inch ³	centimeter ³	16.3871
inch ⁴	centimeter ⁴	41.6231
joule	British thermal unit	0.00095
joule	foot-pound force	0.73756
kilogram	pound mass av.	2.20462
kilogram/meter ³	pound mass/foot ³	0.06243
kilometer/hour	mile/hour statute	0.62137
kilowatt-hour	kilojoule	3,600.0
kilowatt-hour	megajoule	3,600.00
knot	kilometer/hour	1.85200
lambert	candela/meter ²	3,183.1
link	meter	0.20117
meter	foot	3.28084
meter ²	foot ²	10.7639
meter ³	barrel (42 US gal)	6.28981
meter ³	foot ³	35.3147
meter/minute	foot/minute	3.28084
meter/minute	gallon US/minute	264.172
micron	micrometer	1.00000
mil	micrometer	25.4000
mile nautical	kilometer	1.85200
mile US statute	kilometer	1.60934
mile ² US statute	kilometer ²	2.58999
mile (US stat.)/US gal	kilometer/decimeter ³	0.42514
mile (US stat.)/hour	kilometer/hour	1.60934
minute (angle)	radian	0.00029
newton	pound force av.	0.22481
ounce force av	newton	0.27801
ounce mass av	gram	28.3495
ounce mass av	kilogram	0.02835
ounce US fluid	centimeter ³	29.5735
ounce troy	gram	31.1035
part/million (volume)	centimeter ³ /meter ³	1.00000
pascal	pound force/foot ²	0.02089
pascal	pound force/inch ²	0.00015

To convert from	to	Multiply by
pica	centimeter	0.42175
pica	inch	0.16604
pint US dry	decimeter ³	0.55061
pint US liquid	decimeter ³	0.47318
point	centimeter	0.03515
point	inch	0.01384
pound force	newton	4.44822
pound mass av	kilogram	0.45359
pound mass troy	kilogram	0.37324
pound mol	kilomol	0.45359
pound force/foot ²	pascal	47.8803
pound force/inch ²	kilopascal	6.89476
pound force/inch ² /foot	kilopascal/meter ³	22.6206
pound mass/barrel	gram/decimeter ³	2.85301
pound mass/barrel	kilogram/meter ³	2.85301
pound mass/foot ²	kilogram/meter ²	1.48816
pound mass/foot ²	kilogram/meter ²	4.88243
pound mass/foot ³	kilogram/meter ³	16.0185
pound mass/US gallon	gram/centimeter ³	0.11983
pound mass/US gallon	kilogram/decimeter ³	0.11983
pound mass/horsepower-hr.	kilogram/kilowatt-hour	0.60828
pound mass/horsepower-hr.	kilogram/megajoule	0.16897
pound mass/horsepower-hr.	milligram/joule	0.16897
pound mass/hour	tonne/day	0.45359
pound mass/minute	kilogram/minute	0.45359
pound mass/second	kilogram/second	0.45359
pound mass/hour-foot	gram/second-meter	0.41338
pound mass/second-foot	kilogram/second-meter	1.48816
pound mass/second-foot	pascal-second	1.48816
pound mass/second foot ²	kilogram/second-meter ²	4.88243
pound mol	kilomol	0.45359
pound mol/foot ³	kilomol/meter ³	16.0185
pound mol/US gallon	kilomol/meter ³	119.826
pound mol/hour	kilomol/hour	0.45359
pound mol/second	kilomol/second	0.45359
pound force-foot	joule	1.35582
pound force-foot	newton-meter	1.35582
pound force-inch	joule	0.11298
pound force-inch	newton-meter	0.11298
pound force-foot ²	kilogram-meter ²	0.04214
pound force-foot/inch	newton-meter/meter	53.3787
pound force-foot/inch ²	newton-meter/meter	4.44822
pound force-foot/inch ²	joule/centimeter	0.00210
pound force-foot/second	kilogram-meter/second	0.13826
pound force-second/foot ²	pascal-second	47.8803
quart US	decimeter ³	0.94635
radian (angle)	degree	57.2958
radian (angle)	revolution	0.15915
radian (angle)/minute	revolution/minute	0.15915
° Rankin	kelvin	0.55556
revolution	radian (angle)	6.28319
revolution/minute	radian/minute	6.28319
section	hectare	258.999
therm	megajoule	105.506
ton force US	kilonewton	8.89644
ton mass US short	kilogram	907.185
ton mass US short	toone	0.90718
tonne	kilogram	1,000.0
tonne	megagram	1.00000
ton force US/foot ²	megapascal	0.09576
ton force US/inch ²	megapascal	13.7895
ton force US-mile	megajoule	14.3174
ton mass US/day	tonne/hour	0.03780
ton mass US/hour	tonne/day	0.90718
ton mass US/day	kilogram/second	0.01050
ton mass US/hour	kilogram/second	0.25120
ton mass US/minute	kilogram/second	15.1197
ton mass US/year	tonne-annum	0.90718
ton mass US/foot ²	tonne/meter ²	9.76486
torr	pascal	133.322
watt	BTU/minute	0.05687
watt	joule/second	1.00000
watt-hour	joule	3,600.0
watt-second	joule	1.00000
yard	meter	0.91440
yard ²	meter ²	0.83613
yard ³	meter ³	0.76455
year	annum	1.00000

This	times this	gives this
Acre	43,560	sq. ft.
Acre-foot	7,757.8	bbl. (42 gal.)
Ampere-hours	43,560	cu. ft.
Angstroms	3,600	Coulombs
Atm. at 32°F.	0.0001	Microns
"	33.90	ft. of water
"	1.034	g./sq. cm.
"	29.92	in. Hg
"	760.18	mm. Hg
"	14.697	psi
Bbl. (42 gal.)	5.6146	cu. ft.
Bbl./day (42 gal.)	9,702.03	cu. in.
"	1.75	g.p.h.
Bbl./hr. (42 gal.)	0.0292	g.p.m.
"	0.0836	cu. ft./min.
"	0.7	g.p.m.
Board feet	144.0	cu. in.
B.t.u.	778	ft.-lb.
"	252	gram-cal.
"	0.0003927	hp.-hr.
"	0.0002928	kw.-hr.
B.t.u./min.	48,681.68	ft.-lb./hr.
"	778	ft.-lb./min.
"	12.967	ft.-lb./sec.
"	0.02356	hp.
"	0.01758	kw.
B.t.u./sec.	48,681.68	ft.-lb./min.
"	778	ft.-lb./sec.
"	1.4146	hp.
Busheis (U.S. std.)	1,05487	kw.
"	1,2445	cu. ft.
"	2,150	cu. in.
Candles/sq. in.	452.39	ft.-Lamberts
Candlepower (sph.)	0.48695	Lamberts
Carat (intl.)	12.566	lumens
"	3.08647	grains
"	0.2	g.
"	200	mg.
Centimeters	0.03281	ft.
"	0.3937	in.
"	39.37	mls.
Cm. of mercury	0.01316	atm.
"	0.4481	ft. of water
"	27.85	lb./sq. ft.
"	0.1934	psi
Cm. per second	1.9685	ft./min.
"	0.03281	ft./sec.
"	0.02237	m.p.h.
Centipoise	1.0000	g./cm.-sec.
Circular mils	0.00067197	lb./ft.-sec.
Coulombs	0.7854	sq. mls.
"	0.1	Abcoulombs
"	0.0002778	amp.-hr.
Cubic centimeters	0.00003531	cu. ft.
"	0.0002642	gal. (U.S. liq.)
"	0.001	liters
Cubic feet	0.00002298	acre-ft.
"	0.1781	bbl. (42 gal.)
"	0.80356	cu.
"	28.320	cc.
"	1.728	cu. in.
"	7.481	gal. (U.S. liq.)
"	28.32	liters
Cu. ft., million	22.95	acre-ft.
Cu. ft. per min.	10.888	bbl. (42 gal.)/hr.
"	0.1781	bbl. (42 gal.)/min.
"	1.440	cu. ft./day
"	7.481	g.p.m. (U.S. liq.)
Cubic inches	0.000465	bu.
"	16.39	cc.
"	0.0005787	cu. ft.
"	0.004329	gal. (U.S. liq.)
"	0.016387	liters
Degrees of arc	60	minutes
"	0.017453	radians
Dynes	0.0010197	g.
"	0.00002248	lb.
Dyne-cm.	1	ergs
"	0.00000007376	ft.-lb.
"	0.0010197	g.-cm.
"	0.1668	in.
Ems	1	dyne-cm.
Ergs	26.80	amp.-hr.
Faradays	6	ft.
Fathoms	30.48	cm.
Feet	0.18687	fathoms
"	0.000184468	mile (U.S. naut.)
"	0.000189394	mile (U.S. stat.)
"	50.80	cm./sec.
Feet per minute	0.00987	knots/hr.
"	0.01136	m.p.h.
Feet per second	0.68121	m.p.h.
"	0.01136	miles/min.
Feet of water	0.02950	atm.
"	0.4335	psi
Foot-candles	1	lumens/sq. ft.
Foot-Lamberts	0.0022105	candies/sq. in.
"	"	lumens/sq. ft.
Foot-pounds	0.0012853	B.t.u.
"	0.000000505	hp.-hr.
"	1.35582	joules
"	0.0003241	kg.-cal.
"	0.000003768	kw.-hr.
Foot-pounds/hour	0.001284	B.t.u./hr.
"	0.00002141	B.t.u./min.
"	0.00000357	B.t.u./sec.
"	0.01668	ft.-lb./min.
"	0.000000505	hp.
"	0.000003768	kw.
Foot-pounds/min.	0.077118	B.t.u./hr.
"	0.001286	B.t.u./min.
"	0.0000303	hp.
"	0.00002258	kw.
Foot-pounds/sec.	0.001285	B.t.u./sec.
"	0.001818	hp.
"	680	ft.
Furlong		

This	times this	gives this
Gal. (U.S. dry)	0.15558	cu. ft.
"	1.1637	gal. (U.S. liq.)
Gal. (U.S. liq.)	0.02381	bbl. (42 gal.)
"	0.133681	cu. ft.
"	231	cu. in.
"	0.859365	gal. (U.S. dry)
"	3.7854	liters
Gal./hr. (U.S. liq.)	0.1337	cu. ft./hr.
"	0.002228	cu. ft./min.
"	0.01668	g.p.m. (U.S. liq.)
G.p.m. (U.S. liq.)	34.2857	bbl./day (42 gal.)
"	1.4286	bbl./hr. (42 gal.)
"	0.02381	bbl./min. (42 gal.)
"	192.49	cu. ft./day
"	0.1337	cu. ft./min.
"	1.440	gal./day
Grains	0.00014286	lb.-avoird.
Grains/gal. (U.S. liq.)	0.01714	g./l.
"	17.118	p.p.m.
Grams	980.7	dynes
"	0.035274	oz.-avoird.
"	0.0022046	lb.-avoird.
Gram-calories	0.003968	B.t.u.
"	3.0875	ft.-lb.
Grams per liter	58.418	grains/gal.
"	1.000	p.p.m.
"	0.0624	lb./cu. ft.
Horsepower	2,544	B.t.u./hr.
"	42.44	B.t.u./min.
"	33,000	ft.-lb./min.
"	550	ft.-lb./sec.
"	745.70	joules/sec.
"	0.74570	kw.
Horsepower-hours	2,544	B.t.u.
"	1,980,000	ft.-lb.
"	0.7457	kw.-hr.
"	2.54	cm.
Inches	0.08333	ft.
"	25,400.05	microns
"	1.000	mls.
Inches of mercury	0.03242	atm.
"	0.49116	psi
Joules	0.0009488	B.t.u.
"	0.239	calories
"	0.73756	ft.-lb.
Kilograms	2.205	lb.-avoird.
Kilogram-calories	3.9683	B.t.u.
"	3.0875	ft.-lb.
Kilowatts	56.92	B.t.u./min.
"	0.948	B.t.u./sec.
"	44.253	ft.-lb./min.
"	1.341	hp.
Kilowatt-hours	3,415	B.t.u.
"	2,655,199	ft.-lb.
"	1.341	hp.-hr.
"	6,080	ft.
Knots (naut. mile)	1.152	mile (U.S. stat.)
"	101.34	ft./min.
Knots	1.152	m.p.h. (U.S. stat.)
"	1.000	cc.
Liters	0.03531	cu. ft.
"	61.0271	cu. in.
"	0.28418	gal. (U.S. liq.)
"	1.057	qt. (U.S. liq.)
Logarithm, com	2.3026	logarithm, Nap.
Logarithm, Nap	0.4343	logarithm, com
Meters	3.2808	ft.
"	39.37	in.
"	1,000,000	microns
Microns	10,000	angstroms
"	0.00003937	in.
"	0.001	mm.
Mils	0.00254	cm.
"	0.001	in.
"	25.4	microns
Miles (U.S. stat.)	5,280	ft.
"	1.6093	km.
Miles (U.S. naut.)	6,080	ft.
Miles per hour	44.704	cm./sec.
"	88	ft./min.
"	1.4667	ft./sec.
"	0.01668	miles/min.
Milliliters	0.061	cu. in.
Millimeters	0.03937	in.
"	39.37	mls.
Minutes of arc	0.016667	deg. of arc
Ounces (avoird.)	28.35	g.
Parts per million	0.05835	grains/gal.
"	0.001	g./l.
Points	0.013889	in.
Pounds, avoird.	444.823	dynes
"	7.000	grains
"	453.59	g.
"	1.215	lb., Troy, Apoth.
Pounds per gallon	7.4805	lb./cu. ft.
Pounds per cu. ft.	0.018018	g./cc.
"	0.13368	lb./gal. (U.S. liq.)
Pounds per sq. in.	0.06804	atm.
"	5.1715	cm. Hg.
"	2.307	ft. of water
"	2.038	in. Hg.
"	144	lb./sq. ft.
Quarts	0.946	liters
Radians	57.2958	deg. of arc
Radians per sec.	9.549	r.p.m.
Rods	16.5	ft.
"	5.5	yd.
Sections	840	acres
"	1	sq. miles
Square feet	0.00002296	acres
"	144	sq. in.
Square mile	1.2732	circular mile
Township	23,040	acres
"	36	sections
"	36	sq. miles
Vara	2.7777	ft.

GENERAL CONVERSION FACTORS

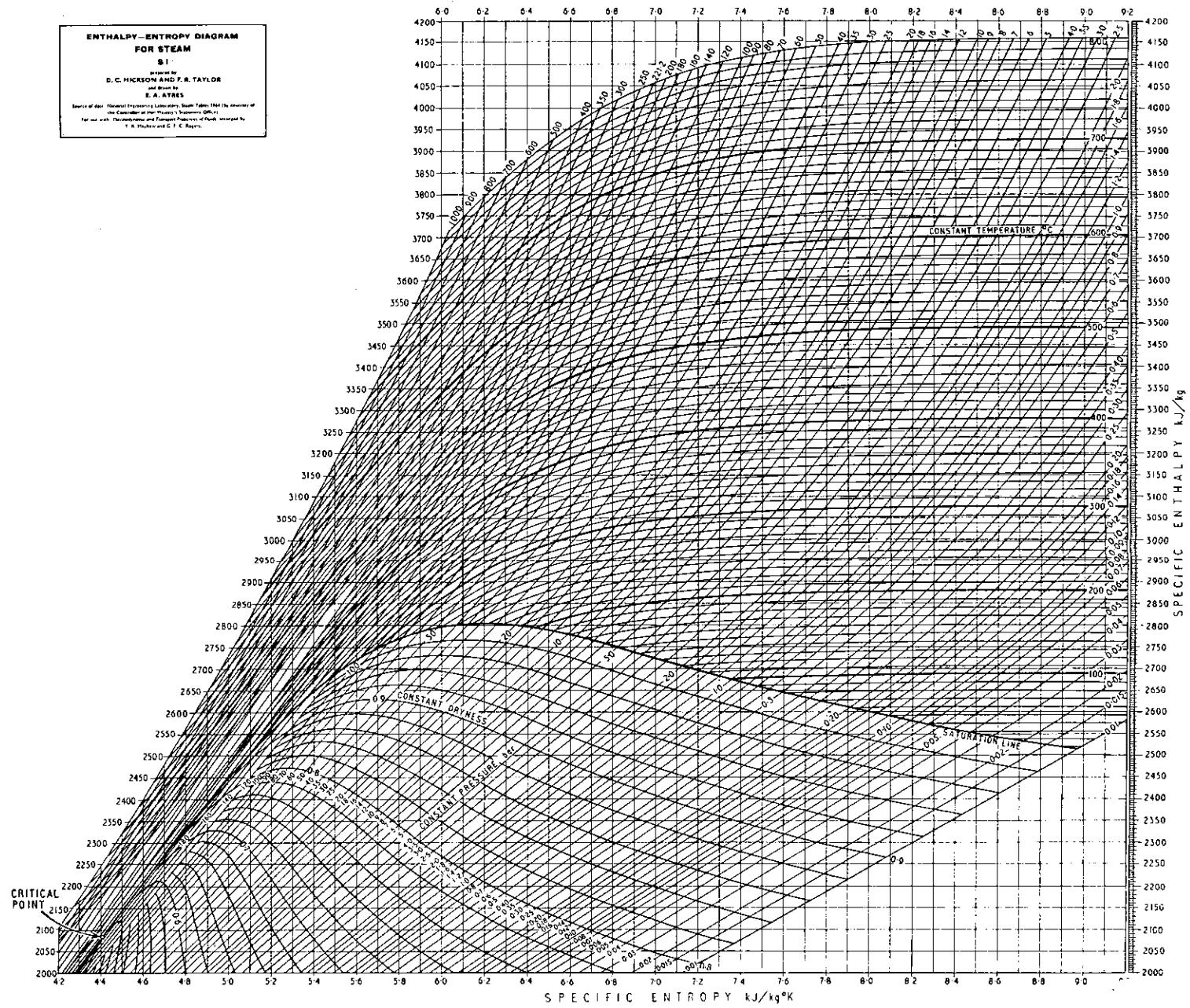
Temp. °C or °F			Temp. °C or °F			Temp. °C or °F			Temp. °C or °F			Temp. °C or °F		
°C	°F	°C	°F	°C	°F	°C	°F	°C	°F	°C	°F	°C	°F	°C
-459.69 to -4			-3 to 83			84 to 800			810 to 1,670			1,680 to 3,000		
-273.16	-459.69	-19.44	-3	28.9	84	183.2	432.2	810	1,490.0	915.6	1,680	3,058.0		
-267.78	-450	-18.89	-2	28.4	29.4	85	185.0	437.8	820	1,508.0	921.1	1,690	3,074.0	
-262.22	-440	-18.33	-1	30.3	30.0	86	186.8	443.3	830	1,526.0	926.7	1,700	3,092.0	
-256.67	-430	-17.8	0	32.0	30.8	87	188.6	448.9	840	1,544.0	932.2	1,710	3,110.0	
-251.11	-420	-17.2	1	33.8	31.1	88	190.4	454.4	850	1,562.0	937.8	1,720	3,128.0	
-245.56	-410	-16.7	2	35.8	31.7	89	192.2	460.0	860	1,580.0	943.3	1,730	3,146.0	
-240.00	-400	-16.1	3	37.4	32.2	90	194.0	465.6	870	1,598.0	948.9	1,740	3,164.0	
-234.44	-390	-15.6	4	39.1	32.8	91	195.8	471.1	880	1,616.0	954.4	1,750	3,182.0	
-228.89	-380	-15.0	5	41.0	33.3	92	197.6	476.7	890	1,634.0	960.0	1,760	3,200.0	
-223.33	-370	-14.4	6	42.8	33.9	93	199.4	482.2	900	1,652.0	965.6	1,770	3,218.0	
-217.78	-360	-13.9	7	44.8	34.4	94	201.2	487.8	910	1,670.0	971.1	1,780	3,236.0	
-212.22	-350	-13.3	8	46.4	35.0	95	203.0	493.3	920	1,688.0	976.7	1,790	3,254.0	
-206.67	-340	-12.8	9	48.2	35.6	96	204.8	498.9	930	1,706.0	982.2	1,800	3,272.0	
-201.11	-330	-12.2	10	50.0	36.1	97	206.6	504.4	940	1,724.0	987.8	1,810	3,290.0	
-195.56	-320	-11.7	11	51.8	36.7	98	208.4	510.0	950	1,742.0	993.3	1,820	3,308.0	
-190.00	-310	-11.1	12	53.8	37.2	99	210.2	515.6	960	1,760.0	998.9	1,830	3,326.0	
-184.44	-300	-10.6	13	55.4	37.8	100	212.0	521.1	970	1,778.0	1,004.4	1,840	3,344.0	
-178.89	-290	-10.0	14	57.2	43.3	110	230.0	526.7	980	1,796.0	1,010.0	1,850	3,362.0	
-173.33	-280	-9.44	15	59.0	48.9	120	248.0	532.2	990	1,814.0	1,015.6	1,860	3,380.0	
-168.89	-273.16	-8.89	16	60.8	54.4	130	266.0	537.8	1,000	1,832.0	1,021.1	1,870	3,398.0	
-168.89	-272	-8.33	17	62.6	60.0	140	284.0	543.3	1,010	1,850.0	1,026.7	1,880	3,416.0	
-167.78	-270	-7.78	18	64.4	65.6	150	302.0	548.9	1,020	1,868.0	1,032.2	1,890	3,434.0	
-162.22	-260	-7.22	19	66.2	71.1	160	320.0	554.4	1,030	1,886.0	1,037.8	1,900	3,452.0	
-156.67	-250	-6.67	20	68.0	76.7	170	338.0	560.0	1,040	1,904.0	1,043.3	1,910	3,470.0	
-151.11	-240	-6.11	21	69.8	82.2	180	356.0	565.6	1,050	1,922.0	1,048.9	1,920	3,488.0	
-145.56	-230	-5.56	22	71.6	87.8	190	374.0	571.1	1,060	1,940.0	1,054.4	1,930	3,506.0	
-140.00	-220	-5.00	23	73.4	93.3	200	392.0	576.7	1,070	1,958.0	1,060.0	1,940	3,524.0	
-134.44	-210	-4.44	24	75.2	98.9	210	410.0	582.2	1,080	1,976.0	1,065.6	1,950	3,542.0	
-128.89	-200	-3.89	25	77.0	104.4	220	428.0	587.8	1,090	1,994.0	1,071.1	1,960	3,560.0	
-123.33	-190	-3.33	26	78.8	110.0	230	446.0	593.3	1,100	2,012.0	1,076.7	1,970	3,578.0	
-117.78	-180	-2.78	27	80.6	115.6	240	464.0	598.9	1,110	2,030.0	1,082.2	1,980	3,596.0	
-112.22	-170	-2.22	28	82.4	121.1	250	482.0	604.4	1,120	2,048.0	1,087.8	1,990	3,614.0	
-106.67	-160	-1.67	29	84.2	126.7	260	500.0	610.0	1,130	2,066.0	1,093.3	2,000	3,632.0	
-101.11	-150	-1.11	30	86.0	132.2	270	518.0	615.6	1,140	2,084.0	1,098.9	2,010	3,650.0	
-95.56	-140	-0.56	31	87.8	137.8	280	536.0	621.1	1,150	2,102.0	1,104.4	2,020	3,668.0	
-90.00	-130	0	32	89.6	143.3	290	554.0	626.7	1,160	2,120.0	1,110.0	2,030	3,686.0	
-84.44	-120	0.56	33	91.4	148.9	300	572.0	632.2	1,170	2,138.0	1,115.6	2,040	3,704.0	
-78.89	-110	1.11	34	93.2	154.4	310	590.0	637.8	1,180	2,156.0	1,121.1	2,050	3,722.0	
-73.33	-100	1.67	35	95.0	160.0	320	608.0	643.3	1,190	2,174.0	1,126.7	2,060	3,740.0	
-70.56	-95	2.22	36	96.8	165.6	330	626.0	648.9	1,200	2,192.0	1,132.2	2,070	3,758.0	
-67.78	-90	2.78	37	98.6	171.1	340	644.0	654.4	1,210	2,210.0	1,137.8	2,080	3,776.0	
-65.00	-85	3.33	38	100.4	176.7	350	662.0	660.0	1,220	2,228.0	1,143.3	2,090	3,794.0	
-62.22	-80	3.89	39	102.2	182.2	360	680.0	665.6	1,230	2,246.0	1,148.9	2,100	3,812.0	
-59.45	-75	4.44	40	104.0	187.8	370	698.0	671.1	1,240	2,264.0	1,154.4	2,110	3,830.0	
-56.67	-70	5.00	41	105.8	193.3	380	716.0	676.7	1,250	2,282.0	1,160.0	2,120	3,848.0	
-53.89	-65	5.56	42	107.6	198.9	390	734.0	682.2	1,260	2,300.0	1,165.6	2,130	3,866.0	
-51.11	-60	6.11	43	109.4	204.4	400	752.0	687.8	1,270	2,318.0	1,171.1	2,140	3,884.0	
-48.34	-55	6.67	44	111.2	210.0	410	770.0	693.3	1,280	2,336.0	1,176.7	2,150	3,902.0	
-45.56	-50	7.22	45	113.0	215.6	420	788.0	698.9	1,290	2,354.0	1,182.2	2,160	3,920.0	
-42.78	-45	7.78	46	114.8	221.1	430	806.0	704.4	1,300	2,372.0	1,187.8	2,170	3,938.0	
-40.00	-40	8.33	47	116.6	226.7	440	824.0	710.0	1,310	2,390.0	1,193.3	2,180	3,956.0	
-39.45	-39	8.89	48	118.4	232.2	450	842.0	715.6	1,320	2,408.0	1,198.9	2,190	3,974.0	
-38.89	-38	9.44	49	120.2	237.8	460	860.0	721.1	1,330	2,426.0	1,204.4	2,200	3,992.0	
-38.34	-37	10.0	50	122.0	243.3	470	878.0	726.7	1,340	2,444.0	1,210.0	2,210	4,010.0	
-37.78	-36	10.6	51	123.8	248.9	480	896.0	732.2	1,350	2,462.0	1,215.6	2,220	4,028.0	
-37.23	-35	11.1	52	125.6	254.4	490	914.0	737.8	1,360	2,480.0	1,221.1	2,230	4,046.0	
-36.67	-34	11.7	53	127.4	260.0	500	932.0	743.3	1,370	2,498.0	1,226.7	2,240	4,064.0	
-36.12	-33	12.2	54	129.2	265.6	510	950.0	748.9	1,380	2,516.0	1,232.2	2,250	4,082.0	
-35.56	-32	12.8	55	131.0	271.1	520	968.0	754.4	1,390	2,534.0	1,237.8	2,260	4,100.0	
-35.00	-31	13.3	56	132.8	276.7	530	986.0	760.0	1,400	2,552.0	1,243.3	2,270	4,118.0	
-34.44	-30	13.9	57	134.6	282.2	540	1,004.0	765.6	1,410	2,570.0	1,248.9	2,280	4,136.0	
-33.89	-29	14.4	58	136.4	287.8	550	1,022.0	771.1	1,420	2,588.0	1,254.4	2,290	4,154.0	
-33.33	-28	15.0	59	138.2	293.3	560	1,040.0	776.7	1,430	2,606.0	1,260.0	2,300	4,172.0	
-32.78	-27	15.6	60	140.0	298.9	570	1,058.0	782.2	1,440	2,624.0	1,265.6	2,310	4,190.0	
-32.22	-26	16.1	61	141.8	304.4	580	1,076.0	787.8	1,450	2,642.0	1,271.1	2,320	4,208.0	
-31.67	-25	16.7	62	143.6	310.0	590	1,094.0	793.3	1,460	2,660.0	1,276.7	2,330	4,226.0	
-31.11	-24	17.2	63	145.4	315.6	600	1,112.0	798.9	1,470	2,678.0	1,282.2	2,340	4,244.0	
-30.56	-23	17.8	64	147.2	321.1	610	1,130.0	804.4	1,480	2,696.0	1,287.8	2,350	4,262.0	
-30.00	-22	18.3	65	149.0	326.7	620	1,148.0	810.0	1,490	2,714.0	1,293.3	2,360	4,280.0	
-29.45	-21	18.9	66	150.8	332.2	630	1,16							

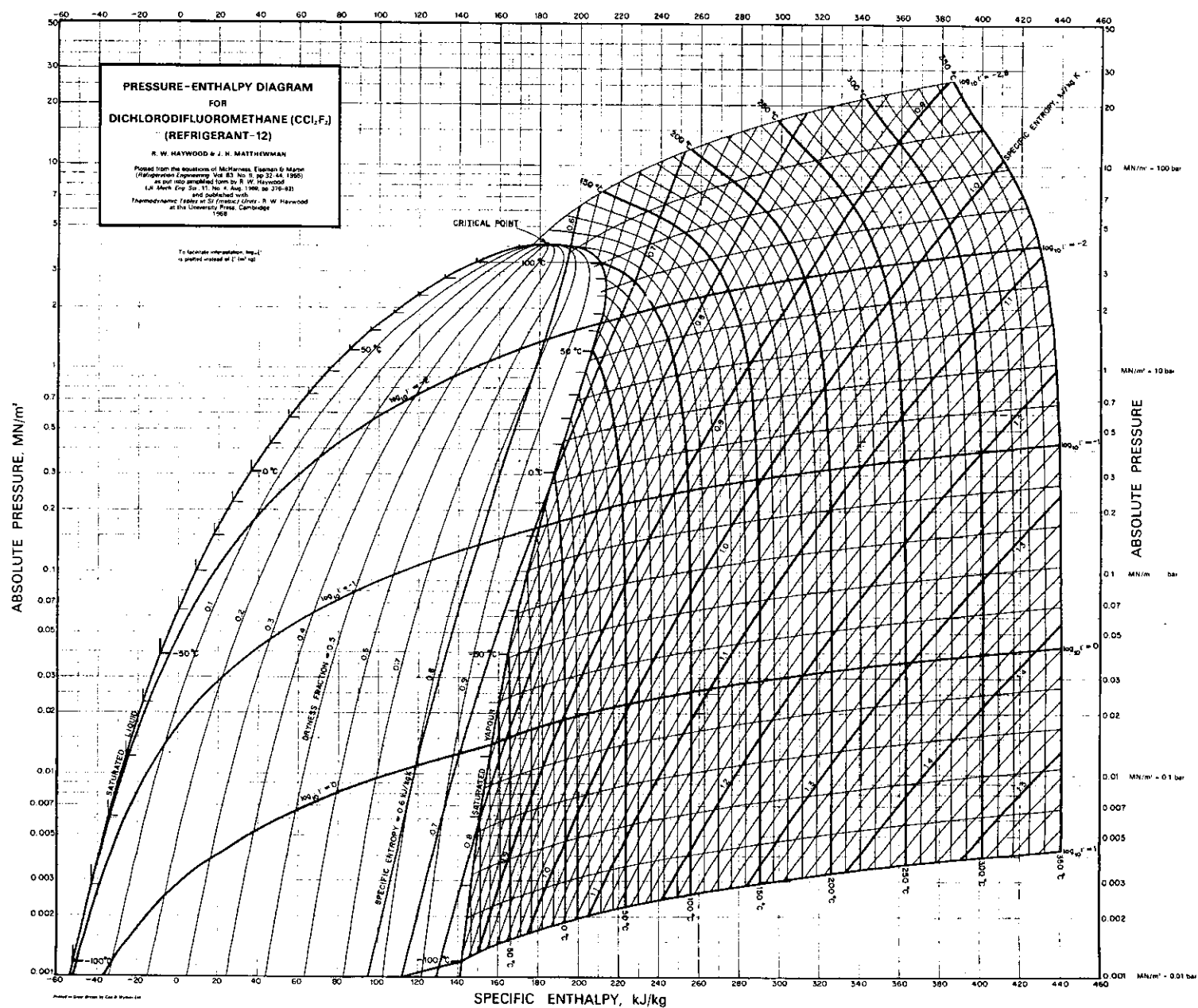
APPENDIX B

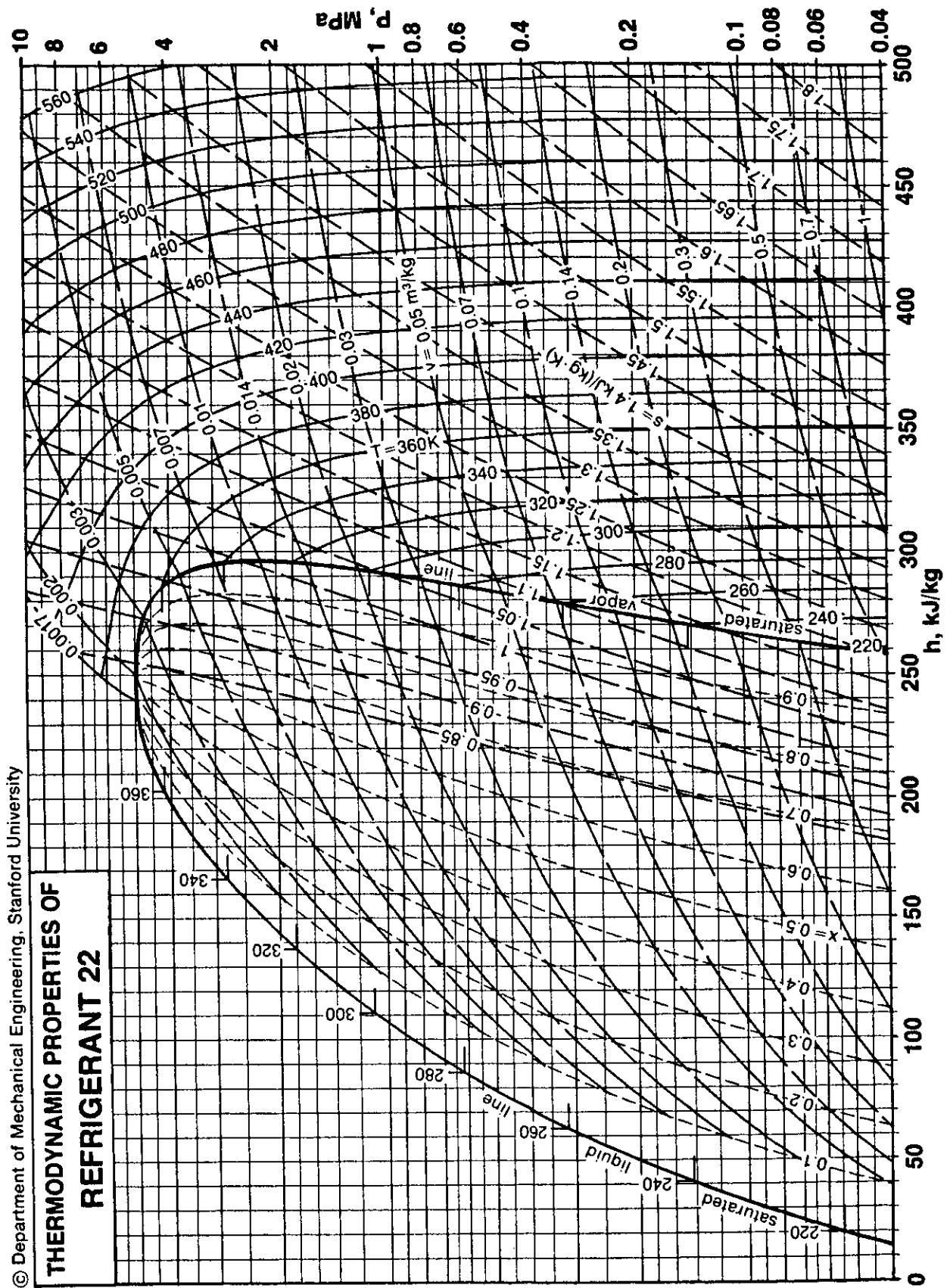
THERMODYNAMIC DATA

Both SI Metric and English unit data are shown. Most is from previous editions. Samples of data from other sources are included.

The H-S diagram for steam, the P-H diagram for Freon 12 and two tables are from the reference shown in the title block. The P-H diagram for Freon 22 is a sample of data from "Thermodynamic Properties in SI" by Reynolds, available from Department of Mechanical Engineering, Stanford University, Stanford, CA 94305.







Ammonia - NH₃ (Refrigerant 717)

Saturation Values							Superheat ($t-t_s$)			
t	p_s	v_g	h_f	h_g	s_f	s_g	50 K		100 K	
							h	s	h	s
-50	0.4089	2.625	-44.4	1373.3	-0.194	6.159	1479.8	6.592	1585.9	6.948
-45	0.5454	2.005	-22.3	1381.6	-0.096	6.057	1489.3	6.486	1596.1	6.839
-40	0.7177	1.552	0	1390.0	0	5.962	1498.6	6.387	1606.3	6.736
-35	0.9322	1.216	22.3	1397.9	0.095	5.872	1507.9	6.293	1616.3	6.639
-30	1.196	0.9633	44.7	1405.6	0.188	5.785	1517.0	6.203	1626.3	6.547
-28	1.317	0.8809	53.6	1408.5	0.224	5.751	1520.7	6.169	1630.3	6.512
-26	1.447	0.8058	62.6	1411.4	0.261	5.718	1524.3	6.135	1634.2	6.477
-24	1.588	0.7389	71.7	1414.3	0.297	5.686	1527.9	6.103	1638.2	6.444
-22	1.740	0.6783	80.8	1417.3	0.333	5.655	1531.4	6.071	1642.2	6.411
-20	1.902	0.6237	89.8	1420.0	0.368	5.623	1534.8	6.039	1646.0	6.379
-18	2.077	0.5743	98.8	1422.7	0.404	5.593	1538.2	6.008	1650.0	6.347
-16	2.265	0.5296	107.9	1425.3	0.440	5.563	1541.7	5.978	1653.8	6.316
-14	2.465	0.4890	117.0	1427.9	0.475	5.533	1545.1	5.948	1657.7	6.286
-12	2.680	0.4521	126.2	1430.5	0.510	5.504	1548.5	5.919	1661.5	6.256
-10	2.908	0.4185	135.4	1433.0	0.544	5.475	1551.7	5.891	1665.3	6.227
-8	3.153	0.3879	144.5	1435.3	0.579	5.447	1554.9	5.863	1669.0	6.199
-6	3.413	0.3599	153.6	1437.6	0.613	5.419	1558.2	5.836	1672.8	6.171
-4	3.691	0.3344	162.8	1439.9	0.647	5.392	1561.4	5.808	1676.4	6.143
-2	3.983	0.3110	172.0	1442.2	0.681	5.365	1564.6	5.782	1680.1	6.116
0	4.295	0.2895	181.2	1444.4	0.715	5.340	1567.8	5.756	1683.9	6.090
2	4.625	0.2699	190.4	1446.5	0.749	5.314	1570.9	5.731	1687.5	6.065
4	4.975	0.2517	199.7	1448.5	0.782	5.288	1574.0	5.706	1691.2	6.040
6	5.346	0.2351	209.1	1450.6	0.816	5.263	1577.0	5.682	1694.9	6.015
8	5.736	0.2198	218.5	1452.5	0.849	5.238	1580.1	5.658	1698.4	5.991
10	6.149	0.2056	227.8	1454.3	0.881	5.213	1583.1	5.634	1702.2	5.967
12	6.585	0.1926	237.2	1456.1	0.914	5.189	1586.0	5.611	1705.7	5.943
14	7.045	0.1805	246.6	1457.8	0.947	5.165	1588.9	5.588	1709.1	5.920
16	7.529	0.1693	256.0	1459.5	0.979	5.141	1591.7	5.565	1712.5	5.898
18	8.035	0.1590	265.5	1461.1	1.012	5.118	1594.4	5.543	1715.9	5.876
20	8.570	0.1494	275.1	1462.6	1.044	5.095	1597.2	5.521	1719.3	5.854
22	9.134	0.1405	284.6	1463.9	1.076	5.072	1600.0	5.499	1722.8	5.832
24	9.722	0.1322	294.1	1465.2	1.108	5.049	1602.7	5.478	1726.3	5.811
26	10.34	0.1245	303.7	1466.5	1.140	5.027	1605.3	5.458	1729.6	5.790
28	10.99	0.1173	313.4	1467.8	1.172	5.005	1608.0	5.437	1732.7	5.770
30	11.67	0.1106	323.1	1468.9	1.204	4.984	1610.5	5.417	1735.9	5.750
32	12.37	0.1044	332.8	1469.9	1.235	4.962	1613.0	5.397	1739.3	5.731
34	13.11	0.0986	342.5	1470.8	1.267	4.940	1615.4	5.378	1742.6	5.711
36	13.89	0.0931	352.3	1471.8	1.298	4.919	1617.8	5.358	1745.7	5.692
38	14.70	0.0880	362.1	1472.6	1.329	4.898	1620.1	5.340	1748.7	5.674
40	15.54	0.0833	371.9	1473.3	1.360	4.877	1622.4	5.321	1751.9	5.655
42	16.42	0.0788	381.8	1473.8	1.391	4.856	1624.6	5.302	1755.0	5.637
44	17.34	0.0746	391.8	1474.2	1.422	4.835	1626.8	5.284	1758.0	5.619
46	18.30	0.0706	401.8	1474.5	1.453	4.814	1629.0	5.266	1761.0	5.602
48	19.29	0.0670	411.9	1474.7	1.484	4.793	1631.1	5.248	1764.0	5.584
50	20.33	0.0635	421.9	1474.7	1.515	4.773	1633.1	5.230	1766.8	5.567

Relative molecular mass = 17.031; further properties of the liquid are given on p.15.

Dichlorodifluoromethane - CF_2Cl_2 (Refrigerant 12)

Saturation Values						Superheat ($t-t_s$)					
t	p_s	v_g	h_f	h_g	s_f	s_g	15 K		30 K		
							h	s	h	s	
-100	0.0118	10.100	-51.84	142.00	-0.2567	0.8628	148.89	0.9019	156.10	0.9428	
-95	0.0181	6.585	-47.56	144.22	-0.2323	0.8442	151.23	0.8830	158.55	0.9195	
-90	0.0284	4.416	-43.28	146.46	-0.2086	0.8274	153.59	0.8649	161.02	0.9010	
-85	0.0424	3.037	-39.00	148.73	-0.1856	0.8122	155.98	0.8493	163.52	0.8851	
-80	0.0617	2.138	-34.72	151.02	-0.1631	0.7985	158.39	0.8351	166.04	0.8706	
-75	0.0879	1.538	-30.43	153.32	-0.1412	0.7861	160.82	0.8226	168.57	0.8578	
-70	0.1227	1.127	-26.13	155.63	-0.1198	0.7749	163.26	0.8110	171.12	0.8459	
-65	0.1680	0.8412	-21.81	157.96	-0.0988	0.7649	165.70	0.8008	173.68	0.8355	
-60	0.2262	0.6379	-17.49	160.29	-0.0783	0.7558	168.15	0.7915	176.26	0.8239	
-55	0.2998	0.4910	-13.14	162.62	-0.0582	0.7475	170.60	0.7830	178.84	0.8172	
-50	0.3915	0.3831	-8.78	164.95	-0.0384	0.7401	173.07	0.7753	181.43	0.8093	
-45	0.5044	0.3027	-4.40	167.28	-0.0190	0.7335	175.54	0.7685	184.01	0.8023	
-40	0.6417	0.2419	0	169.60	0	0.7274	178.00	0.7623	186.60	0.7959	
-35	0.8071	0.1954	4.42	171.90	0.0187	0.7219	180.45	0.7568	189.18	0.7902	
-30	1.004	0.1594	8.86	174.20	0.0371	0.7170	182.90	0.7517	191.76	0.7851	
-25	1.237	0.1312	13.33	176.48	0.0552	0.7127	185.33	0.7473	194.33	0.7805	
-20	1.509	0.1088	17.82	178.73	0.0731	0.7087	187.75	0.7432	196.89	0.7764	
-15	1.826	0.0910	22.33	180.97	0.0906	0.7051	190.15	0.7397	199.44	0.7728	
-10	2.191	0.0766	26.87	183.19	0.1080	0.7020	192.53	0.7365	201.97	0.7695	
-5	2.610	0.0650	31.45	185.38	0.1251	0.6991	194.90	0.7336	204.49	0.7666	
0	3.086	0.0554	36.05	187.53	0.1420	0.6966	197.25	0.7311	206.99	0.7641	
5	3.626	0.0475	40.69	189.66	0.1587	0.6943	199.56	0.7289	209.47	0.7618	
10	4.233	0.0409	45.37	191.74	0.1752	0.6921	201.85	0.7268	211.92	0.7598	
15	4.914	0.0354	50.10	193.78	0.1915	0.6901	204.10	0.7251	214.35	0.7580	
20	5.673	0.0308	54.87	195.78	0.2078	0.6883	206.32	0.7235	216.75	0.7565	
25	6.516	0.0269	59.70	197.73	0.2239	0.6869	208.50	0.7220	219.11	0.7552	
30	7.449	0.0235	64.59	199.62	0.2399	0.6853	210.63	0.7208	221.44	0.7540	
35	8.477	0.0206	69.55	201.45	0.2559	0.6839	212.72	0.7196	223.73	0.7529	
40	9.607	0.0182	74.59	203.20	0.2718	0.6825	214.76	0.7185	225.98	0.7519	
45	10.84	0.0160	79.71	204.87	0.2877	0.6811	216.74	0.7175	228.18	0.7511	
50	12.19	0.0142	84.94	206.45	0.3037	0.6797	218.64	0.7166	230.33	0.7503	
55	13.66	0.0125	90.27	207.92	0.3197	0.6782	220.48	0.7156	232.42	0.7496	
60	15.26	0.0111	95.74	209.26	0.3358	0.6765	222.23	0.7146	234.45	0.7490	
65	16.99	0.00985	101.36	210.46	0.3521	0.6747	223.89	0.7136	236.42	0.7484	
70	18.86	0.00873	107.15	211.48	0.3686	0.6726	225.45	0.7125	238.32	0.7477	
75	20.88	0.00772	113.15	212.29	0.3854	0.6702	226.89	0.7113	240.13	0.7470	
80	23.05	0.00682	119.39	212.83	0.4027	0.6673	228.21	0.7099	241.86	0.7463	
85	25.38	0.00601	125.93	213.04	0.4204	0.6636	229.39	0.7084	243.50	0.7455	
90	27.89	0.00526	132.84	212.80	0.4389	0.6591	230.43	0.7067	245.03	0.7445	
95	30.57	0.00456	140.23	211.94	0.4583	0.6531	231.30	0.7047	246.47	0.7435	
100	33.44	0.00390	148.32	210.12	0.4793	0.6449	231.93	0.7023	247.80	0.7424	
105	36.51	0.00324	157.52	206.57	0.5028	0.6325	232.22	0.6994	248.97	0.7412	
110	39.79	0.00246	169.55	197.99	0.5334	0.6076	232.47	0.6964	250.10	0.7399	
112	41.15	0.00179	183.43	183.43	0.5690	0.5690	232.80	0.6958	250.58	0.7394	

Specific Heat c_p of Some Gases and Vapours

T	CO_2	CO	H_2	N_2	O_2	H_2O	CH_4	C_2H_6	C_3H_8
175	0.709	1.039	13.12	1.039	0.910	1.850	2.083	1.241	
200	0.735	1.039	13.53	1.039	0.910	1.851	2.087	1.260	
225	0.763	1.039	13.83	1.039	0.911	1.852	2.121	1.316	
250	0.791	1.039	14.05	1.039	0.913	1.855	2.156	1.380	1.535
275	0.819	1.040	14.20	1.039	0.915	1.859	2.191	1.453	1.651
300	0.846	1.040	14.31	1.040	0.918	1.864	2.226	1.535	1.766
325	0.871	1.041	14.38	1.040	0.923	1.871	2.293	1.621	1.878
350	0.895	1.043	14.43	1.041	0.928	1.880	2.365	1.709	1.987
375	0.918	1.045	14.46	1.042	0.934	1.890	2.442	1.799	2.095
400	0.939	1.048	14.48	1.044	0.941	1.901	2.525	1.891	2.199
450	0.978	1.054	14.50	1.049	0.956	1.926	2.703	2.063	2.402
500	1.014	1.064	14.51	1.056	0.972	1.954	2.889	2.227	2.596
550	1.046	1.075	14.53	1.065	0.988	1.984	3.074	2.378	2.782
600	1.075	1.087	14.55	1.075	1.003	2.015	3.256	2.519	2.958
650	1.102	1.100	14.57	1.086	1.017	2.047	3.432	2.649	3.126
700	1.126	1.113	14.60	1.098	1.031	2.080	3.602	2.770	3.286
750	1.148	1.126	14.65	1.110	1.043	2.113	3.766	2.883	3.438
800	1.168	1.139	14.71	1.122	1.054	2.147	3.923	2.989	3.581
850	1.187	1.151	14.77	1.134	1.065	2.182	4.072	3.088	3.717
900	1.204	1.163	14.83	1.146	1.074	2.217	4.214	3.180	3.846
950	1.220	1.174	14.90	1.157	1.082	2.252	4.348	3.266	
1000	1.234	1.185	14.98	1.167	1.090	2.288	4.475	3.347	
1050	1.247	1.194	15.06	1.177	1.097	2.323	4.595	3.423	
1100	1.259	1.203	15.15	1.187	1.103	2.358	4.708	3.494	
1150	1.270	1.212	15.25	1.196	1.109	2.392	4.814	3.561	
1200	1.280	1.220	15.34	1.204	1.115	2.425			
1250	1.290	1.227	15.44	1.212	1.120	2.458			
1300	1.298	1.234	15.54	1.219	1.125	2.490			
1350	1.306	1.240	15.65	1.226	1.130	2.521			
1400	1.313	1.246	15.77	1.232	1.134	2.552			
1500	1.326	1.257	16.02	1.244	1.143	2.609			
1600	1.338	1.267	16.23	1.254	1.151	2.662			
1700	1.348	1.275	16.44	1.263	1.158	2.711			
1800	1.356	1.282	16.64	1.271	1.166	2.756			
1900	1.364	1.288	16.83	1.278	1.173	2.798			
2000	1.371	1.294	17.01	1.284	1.181	2.836			
2100	1.377	1.299	17.18	1.290	1.188	2.872			
2200	1.383	1.304	17.35	1.295	1.195	2.904			
2300	1.388	1.308	17.50	1.300	1.202	2.934			
2400	1.393	1.311	17.65	1.304	1.209	2.962			
2500	1.397	1.315	17.80	1.307	1.216	2.987			
2600	1.401	1.318	17.93	1.311	1.223	3.011			
2700	1.404	1.321	18.06	1.314	1.230	3.033			
2800	1.408	1.324	18.17	1.317	1.236	3.053			
2900	1.411	1.326	18.28	1.320	1.243	3.072			
3000	1.414	1.329	18.39	1.323	1.249	3.090			
3500	1.427	1.339	18.91	1.333	1.276	3.163			
4000	1.437	1.346	19.39	1.342	1.299	3.217			
4500	1.446	1.353	19.83	1.349	1.316	3.258			
5000	1.455	1.359	20.23	1.355	1.328	3.292			
5500	1.465	1.365	20.61	1.362	1.337	3.322			
6000	1.476	1.370	20.96	1.369	1.344	3.350			

The specific heats of atomic H, N and O are given with adequate accuracy by $c_p = 2.5 R_0/M$ where M is the relative atomic mass.

T = °C. P = kPa

THE PROPERTIES OF SATURATED STEAM

T	P	v _f	v _g	h _f	h _g	s _f	s _g	T	P	v _f	v _g	h _f	h _g	s _f	s _g
40.00	7.38	1.0092	1955.1	167.37	2574.3	.5731	8.2586	178.00	957.29	1.1239	202.49	753.91	2774.2	2.1191	6.5971
50.00	12.33	1.0166	1204.2	209.19	2591.4	.7054	8.0782	180.00	1001.9	1.1315	194.23	762.62	2774.9	2.1373	6.5832
60.00	19.90	1.0161	767.8	250.89	2609.4	.8306	7.9109	184.00	1098.1	1.1326	177.60	780.42	2779.2	2.1771	6.5495
65.00	25.03	1.0234	6205.6	271.95	2617.7	.8880	7.8282	186.00	1148.6	1.1367	170.24	789.31	2780.3	2.1961	6.5346
70.00	31.05	1.0191	5045.6	292.57	2626.0	.9625	7.7631	189.00	1227.5	1.1399	159.67	802.57	2783.2	2.2250	6.5107
75.00	38.58	1.0271	4133.2	313.81	2635.1	1.0111	7.6791	191.00	1282.7	1.1427	153.09	811.56	2784.3	2.2438	6.4958
80.00	47.32	1.0290	3409.3	334.66	2643.1	1.0753	7.6131	194.00	1368.7	1.1473	143.81	824.94	2786.8	2.2725	6.4724
83.00	53.40	1.0311	3045.1	347.29	2648.2	1.1057	7.5321	205.00	1723.9	1.1643	115.04	874.80	2793.3	2.3763	6.3901
86.00	60.05	1.0344	2728.0	359.83	2653.0	1.1460	7.4798	215.00	2105.4	1.1824	94.646	887.62	2797.5	2.4698	6.3170
90.00	70.04	1.0378	2362.2	376.63	2659.4	1.1927	7.4354	220.00	2319.2	1.1926	86.098	886.19	2798.9	2.5157	6.2810
100.00	101.28	1.0438	1673.0	418.77	2675.3	1.3063	7.3545	230.00	2796.7	1.2063	71.479	909.56	2801.2	2.6089	6.2095
105.00	120.75	1.0482	1419.4	439.83	2683.1	1.3618	7.2958	235.00	3062.1	1.2159	65.284	918.34	2801.4	2.6548	6.1745
110.00	143.21	1.0510	1210.0	460.98	2690.8	1.4178	7.2382	240.00	3345.6	1.2262	59.609	926.1	2801.1	2.6995	6.1361
115.00	169.01	1.0553	1036.3	482.16	2698.2	1.4727	7.1824	245.00	3651.8	1.2373	54.647	931.2	2800.6	2.7481	6.1076
120.00	198.42	1.0616	891.50	503.47	2705.5	1.5270	7.1281	250.00	3975.3	1.2518	50.094	935.0	2798.4	2.7908	6.0655
125.00	232.07	1.0640	770.53	524.54	2712.0	1.5801	7.0764	255.00	4322.1	1.2499	45.859	938.1	2797.7	2.8395	6.0339
130.00	269.98	1.0707	668.10	545.97	2719.5	1.6336	7.0250	260.00	4692.3	1.2903	42.150	941.3	2794.0	2.8820	5.9982
135.00	312.94	1.0749	581.89	567.29	2726.0	1.6859	6.9758	265.00	5084.0	1.2818	38.695	944.4	2791.8	2.9305	5.9619
137.00	331.39	1.0785	551.00	575.91	2729.5	1.7084	6.9555	270.00	5502.5	1.3048	35.579	947.7	2787.5	2.9747	5.9265
140.00	361.40	1.0788	508.83	588.55	2731.1	1.7388	6.9279	275.00	5946.1	1.3141	32.723	950.4	2782.9	3.0204	5.8902
145.00	415.34	1.0852	446.07	610.20	2738.7	1.7895	6.8807	277.00	6131.1	1.3205	31.652	952.7	2780.7	3.0384	5.8757
147.00	438.68	1.0877	423.85	618.73	2740.9	1.8102	6.8620	280.00	6415.0	1.3334	30.113	956.2	2777.7	3.0674	5.8534
150.00	475.94	1.0907	392.32	631.80	2745.1	1.8409	6.8355	285.00	6913.9	1.3499	27.718	962.5	2771.4	3.1128	5.8168
153.00	515.36	1.0942	364.10	644.70	2748.4	1.8712	6.8083	290.00	7440.2	1.3719	25.511	969.2	2764.4	3.1596	5.7795
155.00	543.24	1.0969	346.39	653.39	2751.2	1.8915	6.7907	300.00	8586.3	1.4036	21.618	974.3	2747.1	3.2536	5.7012
157.00	572.06	1.0976	329.92	662.05	2752.2	1.9115	6.7729	305.00	9206.0	1.4262	19.898	979.7	2736.6	3.3012	5.6609
160.00	617.90	1.1023	306.75	675.07	2756.4	1.9415	6.7471	310.00	9863.5	1.4647	18.278	984.7	2721.4	3.3543	5.6139
163.00	666.51	1.1058	285.56	688.12	2759.4	1.9713	6.7213	315.00	10554.1	1.4574	16.843	989.0	2716.1	3.3936	5.5828
165.00	700.56	1.1085	272.40	696.83	2761.5	1.9913	6.7039	335.00	13704.1	1.5943	11.833	1057.7	2643.3	3.6017	5.3869
168.00	754.28	1.1124	253.98	709.93	2764.6	2.0210	6.6787	350.00	16526.2	1.8019	8.7090	1681.4	2548.0	3.7939	5.1863
170.00	792.94	1.1107	242.25	718.97	2767.7	2.0424	6.6620	370.00	21040.0	2.3377	4.8901	1911.4	2320.7	4.1429	4.7793
175.00	892.13	1.1212	216.57	740.62	2771.2	2.0895	6.6217	375.00	22331.1	3.3872	2.6537	2154.7	2033.3	4.5143	4.3262

THE PROPERTIES OF SUPERHEATED STEAM

T	P=6.89kPa(Sat.Temp.=38.74°C)			P=137.8kPa(Sat.Temp.=108.87°C)			P=34.5kPa(Sat.Temp.=72.36°C)			P=68.9kPa(Sat.Temp.=89.56°C)		
	v	h	s	v	h	s	v	h	s	v	h	s
95.0	2462.1	2678.5	8.5966	1399.8	2773.4	7.4611	4902.1	2674.9	7.8456	2437.1	2669.9	7.5148
150.0	2831.1	2783.0	8.8613	1589.4	2882.9	7.6043	5649.1	2781.2	8.1149	2816.8	2778.7	7.7905
205.0	3200.0	2888.7	9.0960	1776.7	2992.1	7.9206	6391.4	2887.6	8.3516	3190.1	2886.2	8.0292
260.0	3568.4	2996.1	9.3085	1962.7	3102.1	8.1171	7130.5	2995.4	8.5545	3561.5	2994.2	8.2434
315.0	3936.7	3105.2	9.5030	2148.1	3213.8	8.2985	7868.4	3104.5	8.7598	3931.1	3103.8	8.4387
370.0	4305.0	3215.9	9.6835	2333.0	3327.0	8.4670	8606.4	3215.4	8.9399	4300.7	3215.0	8.6196
425.0	4673.5	3328.6	9.8516	2517.7	3442.0	8.6262	9343.7	3328.4	9.1085	4669.7	3327.9	8.7882
480.0	5041.1	3443.5	10.009	2702.6	3559.2	8.7757	10080.0	3443.2	9.2667	5038.6	3442.8	8.9454
535.0	5409.5	3560.3	10.160	2886.7	3678.0	8.9185	10817.1	3560.0	9.4167	5407.6	3559.6	9.0964
590.0	5777.8	3679.3	10.301	3088.3	3810.4	9.0663	11554.4	3679.0	9.5589	5776.0	3678.8	9.2386
650.0	6179.7	3811.1	10.450	3457.0	4059.1	9.3211	12357.7	3811.1	9.7068	6177.9	3810.8	9.3866
760.0	6915.8	4059.8	10.704	3825.5	4316.4	9.5579	13834.1	4059.8	9.9612	6915.1	4059.5	9.6414
870.0	7652.4	4317.2	10.941				15305.5	4316.9	10.198	7651.9	4316.7	9.8777
T	P=101.3kPa(Sat.Temp.=100°C)			P=275.7kPa(Sat.Temp.=130.69°C)			P=413.5kPa(Sat.Temp.=144.84°C)			P=551.3kPa(Sat.Temp.=155.57°C)		
	v	h	s	v	h	s	v	h	s	v	h	s
150.0	1911.1	2776.2	7.6085	691.23	2762.4	7.1207	454.59	2750.4	6.9108	388.81	2863.3	7.0325
205.0	2167.6	2884.6	7.8496	789.31	2876.7	7.3745	522.37	2870.0	7.1765	438.24	2979.3	7.2624
260.0	2421.0	2993.3	8.0642	884.48	2987.9	7.5949	587.01	2983.7	7.4014	486.27	3093.0	7.4656
315.0	2673.1	3103.1	8.2603	978.43	3099.1	7.7935	650.30	3096.1	7.6021	533.55	3205.8	7.6503
370.0	2925.3	3214.5	8.4413	1071.7	3211.4	7.9756	712.97	3209.1	7.7860	580.53	3321.4	7.8218
425.0	3176.2	3327.5	8.6098	1164.7	3325.1	8.1455	775.29	3323.3	7.9562	627.21	3437.4	7.9817
480.0	3427.8	3442.5	8.7685	1257.3	3440.4	8.3045	837.27	3439.0	8.1162	677.97	3556.0	8.1459
535.0	3676.7	3559.4	8.9185	1350.0	3557.7	8.4550	899.21	3556.5	8.2666	720.23	3675.0	8.2760
590.0	3929.0	3678.3	9.0610	1442.7	3677.1	8.5975	960.96	3676.0	8.4095	770.80	3807.6	8.4244
650.0	4203.3	3810.6	9.2090	1543.2	3809.5	8.7460	1028.2	3808.5	8.5580	83.38	4057.0	8.6755
760.0	4705.2	4059.3	9.4634	1728.0	4058.4	9.0008	1551.5	4057.7	8.8128	955.78	4314.8	8.9166
870.0	5206.4	4316.6	9.7002	1912.2	4316.0	9.2375	1274.8	4315.3	9.0500			
T	P=689.2kPa(Sat.Temp.=164.34°C)			P=827.0kPa(Sat.Temp.=171.81°C)			P=964.8kPa(Sat.Temp.=178.34°C)			P=1102.7kPa(Sat.Temp.=184.18°C)		
	v	h	s	v	h	s	v	h	s	v	h	s
205.0	308.62	2856.1	6.9183	255.13	2848.7	6.8217	217.82	2841.1	6.7381	188.07	2833.0	6.6632
260.0	348.90	2974.7	7.1531	289.42	2970.2	7.0627	246.84	2965.6	6.9848	214.94	2960.7	6.9162
315.0	387.79	3089.8	7.3589	322.12	3086.5	7.2705	275.17	3083.5	7.1951	240.03	3080.2	7.1294
370.0	425.92	3204.4	7.5448	354.13	3201.9	7.4581	303.64	3199.5	7.3839	264.46	3197.0	7.3195
425.0	463.70	3319.5	7.7167	386.79	3317.6	7.6304	329.92	3315.8	7.5576	288.38	3313.7	7.4939
480.0	501.16	3436.0	7.8770	417.13	3434.3	7.7916	358.48	3432.9	7.7187	312.12	3431.3	7.6660
540.0	541.88	3564.8	8.0416	451.18	3563.4	7.9567	386.37	3562.3	7.8842	337.81	3560.9	7.8215
590.0	575.76	3673.9	8.1717	479.44	3672.7	8.0867	410.65	3671.5	8.0147	359.11	3670.3	7.9523
650.0	616.29	3806.4	8.3206	513.28	3805.5	8.2356	440.71	3804.6	8.1640	384.53	3803.7	8.1016
760.0	690.45	4056.3	8.5762	575.21	4055.6	8.4912	492.87	4054.7	8.4197	431.13	4054.0	8.3577
870.0	764.50	4314.1	8.8134	636.95	4313.7	8.7288	547.86	4312.9	8.6572	477.49	4312.5	8.5953
T	P=1240.5kPa(Sat.Temp.=189.48°C)			P=1378.3kPa(Sat.Temp.=194.33°C)			P=1516.2kPa(Sat.Temp.=198.81°C)			P=1654.0kPa(Sat.Temp.=202.98°C)		
	v	h	s	v	h	s	v	h	s	v	h	s
205.0	166.63	2824.7	6.5951	147.64	2816.1	6.5628	132.89	2807.3	6.4731	120.55	2798.1	6.4171
260.0	190.03	2955.8	6.8550	170.18	2950.9	6.7994	153.88	2945.8	6.7479	140.28	2940.7	6.7007
315.0	212.72	3076.7	7.0711	190.83	3073.4	6.8901	172.86	3070.2	6.9698	157.96	3066.7	6.9254
370.0	234.54	3194.6	7.2625	210.61	3192.0	7.4687	191.04	3189.7	7.1641	174.71	3187.1	7.1209
425.0	255.94	3311.8	7.4373	229.97	3309.9	7.4709	208.72	3308.1	7.3406	191.04	3306.2	7.2983
480.0	277.07	3429.6	7.5999	249.06	3428.2	7.5500	226.17	3426.6	7.5043	207.05	3425.0	7.4624
540.0	300.00	3559.7	7.7658	239.76	3558.3	7.7160	244.98	3557.2	7.6712	224.38	3555.8	7.6298
590.0	318.93	3669.9	7.8970	266.84	3668.2	7.8472	260.60	3667.0	7.8024	238.71	3666.1	7.7613
650.0	341.65	3802.7	8.0464	307.34	3801.8	7.9970	279.21	3800.9	7.9522	255.83	3799.9	7.9112
760.0	383.06	4053.3	8.3028	344.66	4052.6	8.2534	313.20	4051.9	8.2091	286.98	4051.2	8.1684
870.0	424.33	4311.8	8.5405	381.86	4311.3	8.4914	347.08	4310.6	8.4471	318.06	4310.2	8.4065

THERMODYNAMIC DATA

T = °C, P = kPa

THE PROPERTIES OF SATURATED METHANE

T	P	v _f	v _g	h _f	h _g	s _f	s _g	T	P	v _f	v _g	h _f	h _g	s _f	s _g
-170.00	47.43	2.2926	1105.8	11.38	536.39	0.1068	5.1964	-125.00	957.07	2.7421	66.714	175.37	595.01	1.3886	4.2245
-165.00	74.45	2.3290	726.41	28.67	544.99	0.2626	5.0270	-120.00	1199.8	2.8131	53.195	195.01	597.09	1.5164	4.1148
-160.00	114.74	2.3634	488.69	46.75	533.60	0.4093	4.8890	-115.00	1481.9	2.9419	42.800	215.98	598.00	1.6552	4.1617
-155.00	169.41	2.4105	341.06	63.60	561.24	0.5490	4.7611	-110.00	1801.0	2.9119	34.739	238.04	597.51	1.7852	3.8299
-150.00	241.21	2.4538	247.13	81.57	568.99	0.6924	4.6506	-105.00	2179.3	3.5151	26.623	289.82	592.06	1.9521	4.0774
-145.00	330.38	2.5023	183.92	99.70	575.78	0.8382	4.5536	-100.00	2608.1	3.1466	22.776	284.01	592.06	2.0478	3.9673
-140.00	444.52	2.5545	139.21	117.67	581.86	0.9768	4.4631	-95.00	3088.7	3.3192	18.295	313.55	582.64	2.2032	3.7067
-135.00	586.36	2.6109	107.15	136.24	587.37	1.1120	4.3776	-90.00	3632.0	3.7460	14.471	344.42	571.66	2.3681	3.6099
-130.00	755.82	2.6735	84.28	155.63	591.73	1.2511	4.2977	-85.00	4248.0	4.1518	10.702	389.03	546.56	2.5962	3.4361

THE PROPERTIES OF SUPERHEATED METHANE

T	P=68.92kPa (Sat. Temp. = -165.89°C)			P=137.84kPa (Sat. Temp. = -157.67°C)			P=206.75kPa (Sat. Temp. = -152.22°C)			P=275.67kPa (Sat. Temp. = -147.94°C)			P=413.51kPa (Sat. Temp. = -141.22°C)		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
-160.00	827.88	555.99	5.1754	444.03	574.15	4.9740	290.45	570.49	4.7409	234.10	589.77	4.7583	151.14	583.95	4.5113
-150.00	907.30	577.52	5.3579	484.44	595.05	5.1552	317.76	592.79	4.9266	254.55	612.14	4.9332	165.57	607.03	4.6949
-140.00	985.11	598.84	5.5324	523.96	617.64	5.3216	344.27	614.83	5.0974	278.19	633.78	5.0870	179.90	628.56	4.8529
-130.00	1060.3	620.26	5.6957	566.03	637.78	5.4697	376.38	634.32	5.2477	307.56	666.37	5.2977	200.99	662.08	5.0699
-120.00	1145.7	640.09	5.8390	609.12	652.44	5.6741	414.69	667.12	5.4549	325.93	687.46	5.4270	214.87	684.99	5.2024
-110.00	1256.0	672.06	6.0412	651.09	669.97	5.8003	436.91	690.86	5.5821	345.85	708.45	5.5425	228.38	706.80	5.3201
-100.00	1382.1	694.28	6.1663	700.04	713.37	5.9135	463.30	711.62	5.6969	356.29	727.67	5.5971	235.18	717.65	5.3756
-90.00	1446.3	725.58	6.3311	719.98	723.63	5.9672	477.11	721.65	5.7513	366.22	746.88	5.6520	241.94	728.89	5.4311
-80.00	1488.5	736.37	6.3844	739.13	734.33	6.0210	489.80	732.50	5.8055	374.85	765.51	5.7071	247.18	739.18	5.4860
-70.00	1606.0	767.77	6.5374	795.52	766.40	6.1739	528.57	764.88	5.9601	414.00	801.63	5.9053	274.18	782.86	5.6873
-60.00	1667.7	788.85	6.6339	834.03	787.52	6.2712	553.91	786.22	6.0577	433.25	806.74	5.9988	287.17	804.42	5.7824
-50.00	1749.2	810.00	6.7255	872.12	808.84	6.3639	579.33	807.67	6.1508	452.98	828.04	6.0890	300.46	825.05	5.8614
-40.00	1845.1	831.22	6.8161	910.04	830.14	6.4532	604.97	828.94	6.2405	472.08	849.63	6.1769	312.21	846.17	5.9614
-30.00	1902.0	852.95	6.9022	945.98	851.79	6.5408	631.62	850.71	6.3283	492.08	870.43	6.2627	324.83	868.40	6.0880
-20.00	2014.7	884.79	7.0272	1004.8	884.08	6.6656	669.07	883.08	6.4530	500.72	882.08	6.3022	337.45	890.67	6.1687
-10.00	2089.0	906.85	7.1069	1042.2	905.89	6.7456	694.11	904.99	6.5333	519.87	904.07	6.3827	350.31	902.67	6.2466
5.00	2163.7	928.85	7.1841	1080.5	927.89	6.8228	718.77	927.21	6.6109	539.06	925.47	6.4603	363.20	925.08	6.3236
15.00	2241.1	950.73	7.2603	1118.3	950.08	6.8988	745.33	949.23	6.6875	557.28	948.81	6.5362	376.15	947.56	6.4013
25.00	2334.0	985.02	7.3730	1175.7	984.39	7.0116	782.19	983.54	6.8001	585.94	983.09	6.6496	390.77	982.16	6.4369
35.00	2429.4	1007.6	7.4448	1213.4	1006.9	7.0836	808.40	1006.2	6.8726	605.56	1005.7	6.7219	403.38	1004.7	6.5096
45.00	2504.6	1030.7	7.5136	1251.7	1030.0	7.1527	833.41	1029.5	6.9417	624.90	1028.8	6.7914	415.77	1027.9	6.5791
55.00	2579.7	1054.3	7.5816	1290.0	1053.9	7.2206	857.74	1053.2	7.0093	643.40	1052.8	6.8594	428.47	1051.8	6.6483
65.00	2655.5	1077.2	7.6504	1327.6	1076.8	7.2899	884.48	1076.0	7.0782	662.78	1075.5	6.9284	442.72	1074.6	6.7008
75.00	2767.5	1113.9	7.7501	1384.1	1113.5	7.3894	922.15	1113.0	7.1783	691.64	1112.5	7.0386	460.95	1111.6	6.8013
85.00	2842.8	1138.4	7.8157	1421.7	1137.9	7.4553	947.77	1137.4	7.2442	710.17	1137.0	7.0943	472.34	1136.0	6.8920
95.00	2918.0	1163.5	7.8798	1459.4	1163.0	7.5193	973.09	1162.5	7.3083	729.22	1162.1	7.1584	484.20	1161.1	6.9577
105.00	2993.6	1188.7	7.9436	1496.8	1188.4	7.5829	998.16	1187.9	7.3719	748.97	1187.5	7.2227	498.30	1186.7	7.0576
115.00	3106.5	1227.0	8.0385	1553.2	1226.7	7.6779	1035.2	1226.3	7.4569	776.70	1225.8	7.3176	516.83	1225.0	7.1057
125.00	3181.4	1253.1	8.1003	1591.3	1252.6	7.7398	1060.5	1252.2	7.5288	795.49	1251.7	7.3794	529.76	1251.0	7.1674
135.00	3256.2	1279.5	8.1613	1628.7	1279.1	7.8013	1085.6	1278.6	7.5902	814.06	1278.1	7.4404	542.50	1277.4	7.2289
145.00	3331.7	1306.3	8.2222	1665.8	1306.0	7.8623	1110.4	1305.5	7.6512	833.15	1305.0	7.5012	554.79	1304.4	7.2902
155.00	3406.3	1333.6	8.2830	1703.6	1333.4	7.9226	1135.1	1333.3	7.7120	851.98	1332.7	7.5638	567.63	1331.9	7.3512
165.00	3519.6	1375.0	8.3722	1760.3	1374.7	8.0128	1163.4	1374.5	7.8012	880.02	1373.9	7.6517	586.35	1373.5	7.4404
175.00	3594.4	1403.0	8.4322	1797.5	1402.7	8.0718	1198.1	1402.4	7.8611	899.78	1402.0	7.7117	598.74	1401.5	7.5002
185.00	3669.7	1431.2	8.4917	1835.2	1431.0	8.1313	1223.4	1430.5	7.9207	917.90	1430.3	7.7712	611.69	1429.8	7.5597
195.00	3745.0	1460.0	8.5506	1872.8	1459.8	8.1900	1248.8	1459.5	7.9795	936.47	1459.3	7.8300	624.67	1458.9	7.6190
205.00	3857.9	1504.0	8.6382	1929.3	1503.8	8.2782	1285.9	1503.5	8.0671	964.33	1503.3	7.9176	643.24	1502.8	7.7066
215.00	3933.2	1533.5	8.6961	1966.9	1533.3	8.3360	1311.3	1533.1	8.1250	983.50	1532.9	7.9750	655.63	1532.4	7.7646
225.00	4007.9	1563.7	8.7533	2003.9	1563.5	8.3933	1336.0	1563.5	8.1823	1002.0	1563.3	8.0328	667.98	1562.8	7.8218

T	P=551.34kPa (Sat.Temp.=-136.11°C)			P=689.18kPa (Sat.Temp.=-131.94°C)		
	v	h	s	v	h	s
-160.00						
-150.00						
-140.00						
-130.00	120.78	600.79	4.5612	93.878	594.74	4.3656
-120.00	131.26	623.32	4.6783	101.83	618.29	4.5371
-110.00	141.97	657.55	4.9027	116.08	653.63	4.7679
-100.00	159.22	681.31	5.0385	125.68	677.24	4.9072
-95.00	169.60	703.31	5.1588	134.15	699.74	5.0301
-85.00	174.50	714.03	5.2152	138.13	710.84	5.0874
-80.00	179.49	725.15	5.2718	142.39	722.08	5.1447
-70.00	194.44	758.44	5.4315	154.41	755.66	5.3067
-60.00	204.27	780.41	5.5316	162.45	777.93	5.4082
-50.00	214.25	802.33	5.6271	170.37	800.00	5.5048
-40.00	224.27	824.05	5.7188	178.35	821.83	5.5972
-30.00	233.48	845.99	5.8077	186.32	844.50	5.6872
-20.00	248.73	878.94	5.9348	198.25	877.43	5.8146
-10.00	258.42	901.30	6.0158	205.98	899.62	5.8951
5.00	268.00	923.72	6.0942	213.63	922.04	5.9744
15.00	277.22	945.81	6.1717	222.35	944.64	6.0520
25.00	292.41	980.41	6.2880	234.07	979.22	6.1658
35.00	301.95	1003.2	6.3576	241.56	1002.1	6.2388
45.00	311.51	1026.5	6.4272	248.46	1025.4	6.3087
55.00	321.32	1050.4	6.4957	256.67	1049.1	6.3770
65.00	329.78	1073.3	6.5642	263.30	1072.5	6.4465
75.00	344.65	1110.3	6.6553	275.69	1109.5	6.5473
85.00	353.85	1134.9	6.7314	282.55	1134.2	6.6138
95.00	363.88	1160.3	6.7962	290.38	1159.5	6.6786
105.00	374.01	1185.4	6.8604	298.58	1185.0	6.7426
115.00	387.27	1224.1	6.9554	309.49	1223.3	6.8381
125.00	397.73	1250.1	7.0171	317.49	1249.4	6.8999
135.00	407.03	1276.7	7.0786	325.25	1275.8	6.9614
145.00	415.97	1303.8	7.1401	332.73	1302.7	7.0228
155.00	425.71	1330.9	7.2005	341.50	1330.4	7.0837
165.00	440.36	1372.6	7.2903	351.43	1371.7	7.1735
175.00	449.29	1400.9	7.3504	359.44	1400.0	7.2335
185.00	458.90	1429.4	7.4099	367.29	1428.4	7.2931
195.00	468.58	1458.2	7.4687	374.60	1457.4	7.3518
205.00	482.19	1502.4	7.5568	385.55	1501.7	7.4400
215.00	492.48	1532.2	7.6146	392.87	1531.2	7.4978
225.00	501.30	1562.3	7.6723	400.79	1561.6	7.5555

THE PROPERTIES OF SUPERHEATED METHANE (continued)

T	P=2067.55kPa (Sat.Temp.=106.39°C)			P=3445.90kPa (Sat.Temp.=91.61°C)			P=5513.4kPa			P=6891.8kPa			P=10337.7kPa		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
-115.00															
-105.00															
-95.00															
-85.00															
-80.00															
-75.00															
-60.00															
-50.00															
-40.00															
-30.00															
-15.00															
-5.00															
5.00															
15.00															
25.00															
40.00															
50.00															
60.00															
70.00															
80.00															
90.00															
95.00															
105.00															
115.00															
125.00															
140.00															
150.00															
160.00															
170.00															
180.00															
190.00															

h = kJ/kg, v = cm³/g and s = kJ/kg·K

T = °C, P = kPa THE PROPERTIES OF SATURATED PROPANE

T	P	v _f	v _g	h _f	h _g	s _f	s _g	T	P	v _f	v _g	h _f	h _g	s _f	s _g
-62.22	33.94	1.6543	1011.3	378.14	823.25	3.6819	5.7912	15.56	736.73	1.9665	61.429	557.21	912.09	4.4007	5.6275
-61.11	41.35	1.6606	955.15	380.47	824.65	3.6932	5.7870	21.11	856.65	2.0033	53.313	571.40	917.21	4.4481	5.6216
-60.00	43.56	1.6666	911.45	382.79	826.05	3.7041	5.7828	26.67	989.66	2.0408	46.509	585.81	921.86	4.4954	5.6158
-56.67	51.55	1.6731	780.35	389.77	830.23	3.7376	5.7698	32.22	1137.2	2.0782	40.141	600.47	926.28	4.5427	5.6103
-51.11	67.40	1.6874	609.92	401.63	837.21	3.7932	5.7527	37.78	1300.5	2.1163	34.835	615.35	930.70	4.5900	5.6053
-45.56	86.84	1.7062	482.57	413.49	843.72	3.8468	5.7368	43.33	1480.4	2.1550	30.402	630.47	934.65	4.6390	5.6011
-40.00	110.27	1.7249	384.56	425.58	850.47	3.9000	5.7234	48.89	1677.5	2.2049	26.594	646.51	939.07	4.6871	5.5969
-34.44	139.08	1.7442	313.39	438.14	857.21	3.9528	5.7108	54.44	1891.8	2.2549	23.098	663.26	942.79	4.7353	5.5919
-28.89	172.64	1.7642	253.46	450.70	863.95	4.0059	5.6982	60.00	2125.4	2.3111	19.977	680.70	946.51	4.7855	5.5881
-23.33	213.30	1.7848	207.88	463.72	870.70	4.0570	5.6865	65.56	2380.4	2.3829	17.355	698.14	949.30	4.8366	5.5793
-17.78	260.58	1.8060	171.05	476.74	877.21	4.1081	5.6752	71.11	2653.3	2.4734	14.983	717.21	950.70	4.8902	5.5697
-12.22	315.99	1.8291	143.58	490.00	883.72	4.1583	5.6652	76.67	2935.9	2.5795	12.905	738.37	950.23	4.9471	5.5567
-6.67	379.05	1.8541	120.49	503.72	889.77	4.2077	5.6564	82.22	3261.2	2.7262	11.237	761.63	947.91	5.0116	5.5362
-1.11	452.79	1.8797	99.885	516.98	895.58	4.2567	5.6484	86.67	3432.8	2.8985	9.6763	783.49	943.95	5.0685	5.5136
4.44	536.18	1.9072	83.029	530.00	901.16	4.3053	5.6409	87.78	3607.2	2.9416	9.3017	788.84	940.93	5.0828	5.5082
10.00	630.60	1.9359	71.168	543.72	906.74	4.3534	5.6338	93.33	3962.8	3.2525	7.0543	822.09	926.28	5.1749	5.4596

THE PROPERTIES OF SUPERHEATED PROPANE

T	P = 50.65kPa (Sat.Temp.=57.06°C)			P = 84.36kPa (Sat.Temp.=46.11°C)			P = 101.28kPa (Sat.Temp.=42.06°C)			P = 137.84kPa (Sat.Temp.=34.61°C)		
	v	h	s	v	h	s	v	h	s	v	h	s
-50.00	822.56	839.39	5.8130	510.78	851.63	5.7751	422.95	850.70	5.7401	322.20	863.95	5.7372
-40.00	860.88	853.02	5.8711	533.79	865.81	5.8334	442.48	865.16	5.7988	343.56	886.33	5.8241
-30.00	899.16	866.89	5.9293	558.29	882.46	5.9194	471.80	887.05	5.8822	365.08	908.96	5.9104
-15.00	955.54	888.34	6.0152	602.63	910.37	6.0051	500.80	910.01	5.9900	379.56	925.15	5.9669
0.00	1013.4	911.14	6.1011	648.64	941.65	6.1197	539.28	941.72	6.1909	393.79	940.92	6.0236
10.00	1050.6	927.09	6.1585	672.20	960.62	6.1755	558.49	958.89	6.0170	408.14	957.93	6.0800
20.00	1089.2	942.52	6.2154	694.70	974.31	6.2316	578.11	975.81	6.1281	422.44	974.90	6.1354
30.00	1126.8	961.12	6.2730	717.35	993.61	6.2872	596.88	993.13	6.2527	436.63	992.46	6.1904
40.00	1164.4	973.74	6.3294	740.39	1013.7	6.3422	616.10	1011.2	6.3070	450.79	1010.7	6.2454
50.00	1202.9	994.70	6.3847	763.43	1030.3	6.3968	635.36	1029.8	6.3634	464.95	1029.3	6.3005
60.00	1241.1	1014.1	6.4393	786.46	1048.6	6.4511	652.54	1048.9	6.3860	479.11	1048.3	6.3555
70.00	1278.7	1024.1	6.4936	809.52	1069.8	6.5053	669.51	1068.7	6.4262	493.27	1067.7	6.4105
80.00	1316.8	1045.8	6.5478									
90.00	1355.0	1074.5	6.6021									

T	P = 206.75kPa (Sat.Temp.=24.18°C)			P = 275.67kPa (Sat.Temp.=16.17°C)			P = 413.51kPa (Sat.Temp.=4.00°C)			P = 551.34kPa (Sat.Temp.=5.38°C)		
	v	h	s	v	h	s	v	h	s	v	h	s
-15.00	224.94	884.19	5.7428	176.96	905.01	5.7679	119.32	916.82	5.7395	90.822	928.88	5.7316
0.00	239.75	907.14	5.8281	184.58	921.12	5.8257	124.65	933.37	5.8003	95.020	947.11	5.7910
10.00	249.65	923.21	5.8844	192.25	937.56	5.8826	129.92	950.76	5.8591	99.111	964.84	5.8494
20.00	259.52	939.23	5.9413	199.64	954.59	5.9388	135.03	968.71	5.9171	103.06	982.98	5.9073
30.00	269.22	956.20	5.9973	206.95	971.73	5.9956	140.09	986.63	5.9741	107.00	1001.7	5.9654
40.00	278.80	973.74	6.0531	214.25	989.45	6.0520	145.14	1005.1	6.0307	110.94	1021.8	6.0234
50.00	288.35	991.26	6.1089	221.56	1007.9	6.1077	150.20	1024.4	6.0872	114.77	1041.5	6.0809
60.00	297.91	1009.3	6.1646	228.86	1026.9	6.1635	155.26	1044.0	6.1434	118.57	1062.0	6.1375
70.00	307.46	1028.1	6.2204	236.16	1046.3	6.2193	160.25	1064.1	6.1993			
80.00	317.01	1047.3	6.2762	243.47	1066.1	6.2746						
90.00	326.56	1067.0	6.3319									

T	P = 689.18kPa (Sat.Temp.=13.12°C)			P = 895.93kPa (Sat.Temp.=22.89°C)			P = 1102.7kPa (Sat.Temp.=30.95°C)			P = 1309.4kPa (Sat.Temp.=38.06°C)			P = 1516.2kPa (Sat.Temp.=44.36°C)		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
20.00	70.164	923.71	5.6734	53.892	934.03	5.6691	44.155	945.57	5.6760	37.705	959.75	5.6934	30.924	951.05	5.6438
30.00	73.857	942.53	5.7417	57.207	954.07	5.7322	46.886	966.59	5.7319	40.047	981.63	5.7606	33.174	974.88	5.7150
40.00	77.305	960.77	5.7942	60.159	973.72	5.7851	49.358	987.91	5.8058	42.173	1003.1	5.8251	35.205	997.00	5.7817
50.00	80.897	979.58	5.8530	62.802	993.72	5.8561	51.683	1009.0	5.8676	44.171	1025.3	5.8880	37.049	1019.7	5.8470
60.00	84.090	999.07	5.9159	65.386	1014.5	5.9176	53.904	1030.4	5.9295	46.048	1047.2	5.9486	38.786	1042.0	5.9090
70.00	87.050	1018.9	5.9748	67.974	1035.0	5.9780	56.058	1051.9	5.9908						
80.00	90.373	1039.2	6.0328	70.523	1055.9	6.0377									
90.00	93.717	1059.9	6.0918												

Datum - h = 0 and s = 0 for solid at absolute zero

THERMODYNAMIC DATA

T = °C. P = kPa

THE PROPERTIES OF SATURATED AMMONIA

T	P	v _f	v _g	h _f	h _g	s _f	s _g	T	P	v _f	v _g	h _f	h _g	s _f	s _g
-50.00	40.87	1.4246	2625.1	-44.42	1373.0	-0.1943	6.1600	-6.00	340.80	1.5460	359.85	153.54	1437.7	0.6132	5.4210
-45.00	66.05	1.4362	1919.9	-22.14	1338.4	-0.3063	5.9975	-4.00	368.78	1.5521	334.31	162.74	1439.8	0.6473	5.3937
-40.00	71.74	1.4489	1552.0	0.00	1389.8	-0.0000	5.9628	-2.00	398.11	1.5590	310.95	171.95	1441.9	0.6812	5.3666
-35.00	92.43	1.5175	1222.2	18.23	1401.7	0.1105	5.8760	0.00	429.29	1.5663	289.48	181.16	1444.2	0.7151	5.3403
-30.00	119.54	1.4749	963.89	45.21	1405.3	0.1874	5.7856	2.00	462.31	1.5731	269.84	190.35	1446.3	0.7487	5.3142
-28.00	131.59	1.4821	880.53	53.50	1408.3	0.2246	5.7521	4.00	497.26	1.5792	251.72	199.79	1448.4	0.7818	5.2886
-26.00	144.65	1.4860	805.50	62.66	1411.2	0.2605	5.7191	6.00	534.32	1.5864	235.09	208.92	1450.2	0.8152	5.2632
-24.00	158.69	1.4923	738.66	71.77	1414.2	0.2970	5.6870	10.00	614.68	1.6006	205.64	227.67	1454.0	0.8613	5.2138
-22.00	173.82	1.4977	678.42	89.77	1417.8	0.3329	5.6554	15.00	728.03	1.6191	174.81	251.33	1458.4	0.9631	5.1538
-20.00	190.14	1.5033	623.72	98.78	1422.5	0.3684	5.6241	20.00	856.87	1.6387	149.36	274.93	1462.2	1.0443	5.0955
-18.00	207.64	1.5102	574.28	107.75	1425.0	0.4037	5.5937	25.00	1002.07	1.6586	128.28	298.96	1465.7	1.1245	5.0389
-16.00	226.33	1.5151	529.57	116.88	1427.7	0.4397	5.5637	30.00	1165.55	1.6809	110.60	322.97	1468.6	1.2035	4.9837
-14.00	246.50	1.5212	488.95	126.24	1430.4	0.4749	5.5343	35.00	1349.32	1.7000	95.775	347.24	1471.0	1.2820	4.9300
-12.00	267.57	1.5274	452.08	135.17	1432.6	0.5095	5.5049	40.00	1553.84	1.7320	83.283	371.93	1472.8	1.3603	4.8774
-10.00	291.20	1.5335	418.43	144.32	1435.1	0.5444	5.4766	45.00	1780.52	1.7672	72.592	396.60	1474.0	1.4373	4.8249
-8.00	314.68	1.5397	387.86	144.32	1435.1	0.5788	5.4483	50.00	2031.70	1.7748	63.450	422.00	1474.4	1.5147	4.7728

THE PROPERTIES OF SUPERHEATED AMMONIA

P = 34.46 kPa (Sat. Temp. = -52.84°C)				P = 48.24 kPa (Sat. Temp. = -47.16°C)				P = 68.92 kPa (Sat. Temp. = -40.74°C)				P = 96.49 kPa (Sat. Temp. = -34.31°C)			
T	v	h	s	T	v	h	s	T	v	h	s	T	v	h	s
-45.00	3195.1	1385.3	6.2959	2271.6	1382.6	6.1228		-30.00	1693.5	1412.7	6.0784				
-40.00	3268.7	1356.1	6.3426	2325.4	1393.7	6.1709		-25.00	1730.6	1423.6	6.1229	1228.3	1419.9	5.9474	
-35.00	3342.3	1406.7	6.3879	2378.4	1404.6	6.2169		-20.00	1767.8	1434.5	6.1662	1255.4	1431.4	5.9915	
-30.00	3415.4	1417.2	6.4316	2430.7	1415.2	6.2620		-15.00	1804.7	1445.2	6.2085	1282.1	1442.3	6.0354	
-25.00	3488.8	1427.7	6.4753	2484.5	1426.3	6.3044		-10.00	1842.0	1455.9	6.2499	1309.4	1453.2	6.0780	
-20.00	3558.9	1438.1	6.5153	2535.8	1436.0	6.3490		-5.00	1878.9	1466.6	6.2901	1336.0	1464.1	6.1185	
-10.00	3705.0	1459.0	6.5975	2640.6	1457.8	6.4298		0.00	1915.1	1477.3	6.3295	1362.5	1474.9	6.1584	
-5.00	3777.2	1469.6	6.6371	2692.2	1468.4	6.4696		5.00	1952.0	1487.9	6.3682	1388.8	1485.9	6.1979	
0.00	3850.0	1480.0	6.6761	2744.0	1479.1	6.5086		10.00	1988.3	1498.6	6.4062	1415.2	1496.7	6.2367	
5.00	3921.3	1490.5	6.7139	2795.5	1489.5	6.5469		15.00	2024.2	1509.3	6.4436	1441.5	1507.3	6.2745	
10.00	3992.9	1501.2	6.7512	2847.3	1500.0	6.5846		20.00	2060.5	1519.8	6.4802	1468.0	1518.3	6.3111	
15.00	4065.1	1511.7	6.7881	2899.3	1510.6	6.6215		25.00	2097.6	1531.1	6.5163	1492.9	1529.2	6.3481	
20.00	4135.9	1521.9	6.8245	2949.9	1521.5	6.6578		30.00	2132.1	1540.6	6.5524	1519.9	1539.4	6.3843	
25.00	4208.2	1532.9	6.8599	3012.0	1531.9	6.6940		35.00	2168.6	1553.1	6.5867	1545.7	1551.0	6.4185	
30.00	4279.6	1543.4	6.8953	3053.1	1542.0	6.7283		40.00	2207.6	1561.8	6.6226	1571.8	1561.8	6.4543	
35.00	4350.9	1553.1	6.9306	3104.2	1554.0	6.7656		50.00	2277.5	1584.4	6.6897	1623.3	1583.2	6.5222	
40.00	4422.3	1565.7	6.9633	3155.6	1562.8	6.7978		60.00	2313.1	1595.2	6.7231	1649.0	1593.9	6.5556	
50.00	4565.0	1586.0	7.0319	3257.7	1585.4	6.8653		70.00	2348.5	1606.1	6.7558	1674.9	1604.8	6.5888	
55.00	4636.4	1596.8	7.0646	3308.6	1596.1	6.8992		80.00	2384.7	1616.8	6.7883	1700.9	1616.0	6.6217	
60.00	4707.7	1607.4	7.0975	3359.9	1607.0	6.9321		90.00	2420.4	1628.2	6.8207	1726.6	1627.0	6.6538	
65.00	4779.0	1618.3	7.1299	3411.0	1617.8	6.9645		100.00	2456.5	1638.9	6.8520	1751.6	1637.9	6.6854	
70.00	4850.4	1629.2	7.1616	3461.6	1628.8	6.9963									
75.00	4921.4	1640.1	7.1932	3512.5	1639.6	7.0279									
80.00	4992.2	1651.1	7.2249	3564.0	1650.6	7.0595									

P = 124.05 kPa (Sat. Temp. = -29.23°C)				P = 165.40 kPa (Sat. Temp. = -23.10°C)				P = 206.75 kPa (Sat. Temp. = -18.09°C)				P = 261.89 kPa (Sat. Temp. = -12.54°C)			
T	v	h	s	T	v	h	s	T	v	h	s	T	v	h	s
-30.00				-15.00	737.68	1434.7	5.7490	585.00	1430.0	5.6246		-10.00	467.30	1435.8	5.5374
-25.00	949.20	1416.4	5.8128	-10.00	753.63	1446.3	5.7937	598.60	1442.0	5.6717		-5.00	478.31	1448.5	5.5843
-20.00	970.69	1427.9	5.8591	-5.00	770.52	1457.8	5.8368	611.94	1454.0	5.7160		0.00	488.94	1460.5	5.6293
-15.00	991.84	1439.4	5.9038	0.00	786.19	1469.3	5.8787	624.92	1465.5	5.7594		5.00	499.55	1472.5	5.6729
-10.00	1013.5	1450.5	5.9463	5.00	802.09	1480.5	5.9196	638.40	1477.1	5.8011		10.00	510.04	1484.4	5.7150
-5.00	1034.5	1461.6	5.9887	10.00	818.43	1491.6	5.9595	651.12	1488.6	5.8416		15.00	520.33	1496.0	5.7557
0.00	1055.4	1472.8	6.0295	15.00	834.25	1502.7	5.9983	663.16	1499.8	5.8820		20.00	530.53	1507.4	5.7954
5.00	1075.9	1483.8	6.0694	20.00	849.48	1514.0	6.0365	676.93	1511.4	5.9202		25.00	540.75	1519.0	5.8346
10.00	1096.9	1494.7	6.1085	25.00	864.98	1524.6	6.0742	688.28	1522.1	5.9588		30.00	550.89	1530.3	5.8713
15.00	1117.9	1505.6	6.1466	30.00	879.96	1536.1	6.1103	701.81	1533.9	5.9952		40.00	570.86	1552.9	5.9457
20.00	1137.6	1516.3	6.1842	35.00	897.90	1547.5	6.1479	714.17	1544.2	6.0322		45.00	580.72	1564.2	5.9814
25.00	1158.8	1527.5	6.2209	40.00	909.94	1557.1	6.1811	726.53	1556.8	6.0679		50.00	590.71	1575.5	6.0167
30.00	1179.0	1538.4	6.2565	50.00	942.29	1580.1	6.2517	751.24	1578.1	6.1379		55.00	600.44	1586.9	6.0512
35.00	1199.2	1549.6	6.2942	60.00	973.63	1602.7	6.3191	775.98	1589.4	6.1723		60.00	610.29	1598.1	6.0851
40.00	1219.7	1559.3	6.3259	65.00	987.80	1613.1	6.3520	788.15	1611.5	6.2391		65.00	620.29	1609.4	6.1184
50.00	1280.0	1592.9	6.4300	70.00	1003.0	1624.5	6.3844	800.90	1622.7	6.2726		70.00	629.31	1620.7	6.1525
55.00	1300.4	1604.0	6.4636	75.00	1018.1	1635.1	6.4165	812.79	1633.7	6.3037		75.00	639.87	1632.1	6.1837
60.00	1320.7	1614.8	6.4966	80.00	1033.7	1646.9	6.4492	824.25	1645.5	6.3370		80.00	649.19	1642.8	6.2175
65.00	1340.6	1626.0	6.5281	90.00	1063.7	1669.0	6.5108	849.28	1667.6	6.3992		85.00	659.30	1654.8	6.2484
70.00	1360.3	1637.0	6.5607	95.00	1078.7	1680.3	6.5416	861.09	1678.9	6.4303		95.00	678.22	1677.3	6.3111
75.00	1380.8	1648.1	6.5920	105.00	1108.5	1702.9	6.6021	885.34	1701.5	6.4908		105.00	697.76	1700.1	6.3723
80.00	1400.6	1659.2	6.6233	115.00	1139.0	1725.7	6.6617	909.51	1724.3	6.5503		115.00	716.86	1723.9	6.4322
85.00	1420.3	1670.3	6.6545					933.39	1747.4	6.6091		125.00	735.96	1745.9	6.4908
90.00	1462.0	1692.6	6.7151									135.00	755.07	1769.1	6.5482
100.00												145.00			

h = kJ/kg, v = cm³/g and s = kJ/kg·K

Datum - Sat. liquid h and s at -40°C.

THE PROPERTIES OF SUPERHEATED AMMONIA (continued)

P = 330.81kPa (Sat.Temp. = -6.78°C)				P = 413.51kPa (Sat.Temp. = -0.99°C)				P = 551.3kPa (Sat.Temp. = -6.89°C)				P = 689.2kPa (Sat.Temp. = 13.36°C)			
T	v	h	s	T	v	h	s	T	v	h	s	T	v	h	s
-10.00				5.00	308.77	1459.1	5.4117	10.00	231.73	1459.8	5.2833				
-5.00	373.66	1441.2	5.4488	10.00	315.89	1471.9	5.4575	15.00	237.39	1473.5	5.3308	185.84	1461.9	5.1897	
0.00	382.49	1453.9	5.4965	15.00	322.86	1484.7	5.5014	20.00	242.78	1487.1	5.3770	190.47	1476.0	5.2391	
5.00	391.20	1466.6	5.5420	20.00	329.75	1496.9	5.5450	25.00	248.27	1499.4	5.4205	195.08	1489.8	5.2863	
10.00	399.79	1478.8	5.5856	25.00	336.69	1509.5	5.5849	30.00	253.51	1512.8	5.4630	199.57	1503.6	5.3299	
15.00	408.22	1490.7	5.6280	30.00	343.20	1521.1	5.6265	35.00	258.85	1526.7	5.5037	204.17	1517.5	5.3744	
20.00	416.56	1502.8	5.6686	35.00	350.00	1533.0	5.6660	40.00	264.35	1539.7	5.5426	208.87	1531.4	5.4199	
25.00	424.74	1514.3	5.7092	40.00	356.54	1545.0	5.7031	45.00	269.95	1549.7	5.5826	213.41	1545.1	5.4658	
30.00	433.06	1526.3	5.7462	45.00	363.08	1556.7	5.7402	50.00	275.98	1561.7	5.6203	218.03	1558.9	5.5117	
35.00	441.16	1539.3	5.7823	50.00	369.57	1568.4	5.7767	55.00	282.92	1573.7	5.6571	222.64	1572.7	5.5576	
40.00	449.16	1550.7	5.8223	55.00	376.04	1580.2	5.8122	60.00	289.81	1585.6	5.6932	227.24	1586.5	5.6035	
45.00	457.10	1560.7	5.8592	60.00	382.43	1591.6	5.8473	65.00	296.69	1597.5	5.7286	231.82	1599.7	5.6494	
50.00	465.05	1572.5	5.8948	65.00	388.77	1603.1	5.8820	70.00	303.41	1609.5	5.7637	236.40	1612.9	5.6953	
55.00	472.95	1583.7	5.9299	70.00	395.17	1614.9	5.9160	75.00	310.41	1621.3	5.7970	240.98	1626.0	5.7412	
60.00	480.82	1595.1	5.9645	75.00	401.41	1626.5	5.9496	80.00	317.66	1632.5	5.8322	245.56	1639.1	5.7871	
65.00	488.68	1606.8	5.9884	80.00	407.72	1637.6	5.9819	85.00	324.92	1643.8	5.8671	250.14	1652.2	5.8330	
70.00	496.50	1617.8	6.0320	85.00	414.00	1648.2	6.0139	90.00	332.18	1655.0	5.9023	254.72	1665.3	5.8789	
75.00	504.29	1629.6	6.0652	90.00	420.22	1660.8	6.0469	95.00	339.44	1666.2	5.9371	259.30	1678.4	5.9248	
80.00	511.99	1640.8	6.0972	95.00	426.47	1672.6	6.0784	100.00	346.70	1680.0	5.9680	263.88	1691.5	5.9707	
85.00	519.79	1652.1	6.1301	100.00	432.69	1684.2	6.1096	105.00	353.96	1691.8	5.9923	268.46	1704.6	6.0166	
90.00	527.59	1663.4	6.1607	105.00	438.87	1695.7	6.1404	110.00	361.22	1703.5	6.0231	273.04	1717.7	6.0625	
95.00	535.19	1675.2	6.1926	110.00	445.17	1707.9	6.1712	115.00	368.48	1715.2	6.0537	277.62	1730.8	6.1084	
100.00	542.79	1687.0	6.2245	115.00	451.17	1718.9	6.2011	120.00	375.74	1727.1	6.0839	282.20	1743.9	6.1543	
105.00	550.33	1698.0	6.2543	120.00	457.48	1729.4	6.2304	125.00	383.00	1738.9	6.1137	286.78	1757.0	6.2002	
110.00	557.81	1710.0	6.2843	125.00	463.87	1740.4	6.2596	130.00	390.26	1748.8	6.1431	291.36	1770.1	6.2461	
115.00	565.81	1721.0	6.3146	130.00	470.44	1751.9	6.2887	135.00	397.52	1759.2	6.1725	295.94	1783.2	6.2920	
120.00	573.81	1732.0	6.3449	135.00	477.44	1763.9	6.3178	140.00	404.78	1769.7	6.2013	300.52	1796.3	6.3379	
125.00	581.81	1743.0	6.3752	140.00	484.94	1776.4	6.3479	145.00	412.04	1781.7	6.2299	305.10	1809.4	6.3838	
130.00	589.26	1754.0	6.4055												
135.00	596.71	1765.0	6.4358												
140.00	604.16	1776.0	6.4661												
145.00	611.45	1787.0	6.4964												

P = 964.9kPa (Sat.Temp. = 23.77°C)				P = 1102.7kPa (Sat.Temp. = 28.13°C)				P = 1240.5kPa (Sat.Temp. = 32.10°C)				P = 1378.4kPa (Sat.Temp. = 35.74°C)			
T	v	h	s	T	v	h	s	T	v	h	s	T	v	h	s
30.00	137.57	1483.9	5.1165	121.21	1489.0	5.0748		105.76	1479.3	4.9922		45.00	98.650	1501.5	5.0186
35.00	140.96	1498.6	5.1641	124.27	1504.1	5.1210		108.66	1495.2	5.0430		50.00	101.22	1517.2	5.0669
40.00	144.33	1512.6	5.2105	127.30	1518.8	5.1640		111.38	1510.3	5.0917		55.00	103.63	1532.1	5.1129
45.00	147.59	1526.9	5.2539	130.16	1532.8	5.1798		114.11	1525.2	5.1375		60.00	106.00	1546.5	5.1569
50.00	150.78	1540.4	5.2965	132.97	1546.7	5.2075		116.70	1539.5	5.1814		65.00	108.38	1560.8	5.1991
55.00	153.91	1553.6	5.3377	135.78	1560.2	5.2391		119.24	1553.5	5.2239		70.00	110.63	1574.5	5.2401
60.00	157.01	1566.7	5.3771	138.54	1573.4	5.2717		121.77	1567.3	5.2649		75.00	112.82	1588.7	5.2800
65.00	160.06	1579.7	5.4153	141.23	1586.5	5.3052		124.22	1580.8	5.3051		80.00	115.17	1603.4	5.3171
70.00	162.99	1592.5	5.4538	143.88	1599.9	5.3433		126.74	1594.0	5.3427		85.00	117.16	1618.2	5.3568
75.00	166.03	1605.3	5.4897	146.57	1612.3	5.3819		129.25	1607.1	5.3816		90.00	119.16	1633.0	5.3976
80.00	169.01	1617.3	5.5255	149.21	1624.7	5.4200		131.73	1620.2	5.4200		95.00	121.50	1647.8	5.4394
85.00	172.00	1629.3	5.5613	151.89	1637.1	5.4587		134.18	1633.2	5.4584		100.00	123.56	1662.6	5.4826
90.00	175.00	1641.3	5.5971	154.51	1649.4	5.4974		136.60	1645.8	5.4982		105.00	125.61	1677.4	5.5268
95.00	178.00	1653.3	5.6329	157.13	1661.7	5.5361		139.00	1658.1	5.5361		110.00	127.73	1692.2	5.5710
100.00	181.00	1665.3	5.6687	159.75	1674.0	5.5748		141.38	1670.4	5.5748		115.00	129.77	1707.0	5.6152
105.00	184.00	1677.3	5.7045	162.37	1686.3	5.6135		143.75	1682.6	5.6135		120.00	131.71	1721.8	5.6594
110.00	187.00	1689.3	5.7403	165.00	1698.6	5.6522		146.11	1694.9	5.6522		125.00	133.89	1736.6	5.7036
115.00	190.00	1701.3	5.7761	167.62	1710.9	5.6909		148.48	1707.2	5.6909		130.00	136.00	1751.4	5.7478
120.00	193.00	1713.3	5.8119	170.25	1723.2	5.7296		150.84	1719.5	5.7296		135.00	138.12	1766.2	5.7920
125.00	196.00	1725.3	5.8477	172.87	1735.5	5.7683		153.19	1731.8	5.7683		140.00	140.14	1781.0	5.8362
130.00	199.00	1737.3	5.8835	175.50	1747.8	5.8070		155.55	1744.1	5.8070		145.00	142.16	1795.8	5.8804
135.00	202.00	1749.3	5.9193	178.12	1760.1	5.8457		157.90	1756.4	5.8457		150.00	144.18	1810.6	5.9246
140.00	205.00	1761.3	5.9551	180.75	1772.4	5.8844		160.26	1768.7	5.8844		155.00	146.19	1825.4	5.9688
145.00	208.00	1773.3	5.9909	183.37	1784.7	5.9231		162.61	1781.0	5.9231		160.00	148.21	1840.2	6.0130
150.00	211.00	1785.3	6.0267	186.00	1797.0	5.9618		164.97	1793.3	5.9618		165.00	150.22	1855.0	6.0572
155.00	214.00	1797.3	6.0625	188.62	1809.3	6.0005		167.32	1805.6	6.0005		170.00	152.24	1869.8	6.1014
160.00	217.00	1809.3	6.0983	191.25	1821.6	6.0392		169.68	1817.9	6.0392					
165.00	220.00	1821.3	6.1341	193.87	1833.9	6.0779		172.03	1830.2	6.0779					
170.00	223.00	1833.3	6.1699	196.50	1846.2	6.1166		174.39	1842.5	6.1166					

P = 1516.2kPa (Sat.Temp. = 39.12°C)				P = 1654.0kPa (Sat.Temp. = 42.27°C)				P = 1791.9kPa (Sat.Temp. = 45.23°C)				P = 2067.5kPa (Sat.Temp. = 50.67°C)			
T	v	h	s	T	v	h	s	T	v	h	s	T	v	h	s
45.00	88.195	1492.6	4.9502	79.492	1483.3	4.8840		74.257	1491.2	4.8759		64.097	1490.6	4.8157	
50.00	90.633	1508.9	4.9999	81.778	1500.3	4.9368		76.403	1508.4	4.9280		66.049	1508.6	4.8701	
55.00	92.947	1524.5	5.0486	84.032	1516.4	4.9869		78.4727							

THERMODYNAMIC DATA

T = °C. P = kPa THE PROPERTIES OF SATURATED CARBON DIOXIDE

T	P	v_f	v_g	h_f	h_g	s_f	s_g	T	P	v_f	v_g	h_f	h_g	s_f	s_g
-95.00	23.42	0.6298	1444.6	-282.22	300.71	2.5426	5.8260	-30.00	1426.7	0.9308	27.002	18.76	322.28	3.9394	5.1878
-85.00	59.83	0.6393	602.03	-276.55	306.48	2.6091	5.7275	-25.00	1662.0	0.9503	22.884	29.52	322.26	3.9792	5.1587
-75.00	133.70	0.6412	268.80	-255.91	311.74	2.6683	5.5466	-20.00	1967.2	0.9708	19.470	38.07	322.33	4.0122	5.1248
-65.00	198.35	0.6479	185.64	-252.44	313.47	2.7049	5.4844	-15.00	2289.4	0.9906	16.599	50.82	324.25	4.0662	5.1246
-55.00	286.98	0.6511	128.85	-242.60	314.98	2.7425	5.4235	-10.00	3044.4	1.0478	12.144	72.56	321.53	4.1426	5.0713
-45.00	410.05	0.6576	91.451	-235.69	315.89	2.7854	5.3694	0.00	3493.7	1.0817	10.375	85.10	319.87	4.1864	5.0451
-35.00	517.57	0.8490	72.229	-31.86	316.05	3.7200	5.3273	5.00	3970.1	1.1321	8.847	99.37	317.11	4.2323	5.0168
-25.00	682.63	0.8667	55.411	-19.49	317.88	3.7767	5.2883	10.00	4504.5	1.1662	7.5225	112.56	313.95	4.2781	4.9894
-15.00	632.56	0.8815	45.813	-9.79	319.25	3.8187	5.2610	15.00	5090.7	1.2203	6.3318	127.16	308.13	4.3284	4.9550
-5.00	1004.2	0.8971	38.162	0.00	320.47	3.8594	5.2348	20.00	5731.7	1.3032	5.2465	144.45	298.89	4.3840	4.9171
5.00	1201.8	0.9133	32.001	9.48	321.21	3.8996	5.2080	25.00	6428.7	1.3964	4.1991	162.56	284.83	4.4457	4.8688
								30.00	7202.0	1.9099	2.6387	207.36	242.38	4.5908	4.7116

THE PROPERTIES OF SUPERHEATED CARBON DIOXIDE

T	P = 6.89kPa			P = 68.92kPa			P = 137.84kPa			P = 275.67kPa			P = 551.34kPa		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
-55.00	5983.8	662.13	6.2013	591.68	660.86	5.7662	292.31	659.45	5.6341	143.00	656.63	5.4967			
-45.00	6258.0	670.24	6.2369	619.74	669.07	5.8019	307.20	667.92	5.6700	150.49	665.35	5.5331	72.273	660.48	5.3521
-15.00	7080.6	694.75	6.3359	703.78	693.61	5.9009	348.89	692.42	5.7695	173.10	690.20	5.6340	84.524	685.55	5.4723
10.00	7766.0	715.58	6.4125	772.85	714.65	5.9775	382.00	713.49	5.8464	190.59	711.40	5.7116	93.517	707.21	5.5617
35.00	8451.4	737.15	6.4843	841.87	736.44	6.0493	419.03	735.51	5.9183	207.74	733.52	5.7842	103.23	730.19	5.6397
65.00	9274.1	763.23	6.5651	924.91	762.76	6.1301	461.65	762.05	5.9991	229.86	760.89	5.8659	113.93	758.53	5.7256
90.00	9958.8	785.55	6.6286	993.42	785.31	6.1936	495.77	784.87	6.0625	246.91	783.88	5.9302	122.52	782.31	5.7922
150.00	11604.1	841.52	6.7704	1158.6	841.28	6.3354	578.79	840.83	6.2044	288.86	840.36	6.0729	143.85	838.75	5.9386
205.00	13106.1	895.21	6.8889	1310.0	894.98	6.4539	654.39	894.75	6.3228	326.90	894.29	6.1922	163.00	893.35	6.0586
315.00	16122.1	1009.6	7.1036	1612.2	1009.6	6.6866	805.81	1009.4	6.5375	402.81	1009.2	6.4069	201.20	1008.7	6.2754
425.00	19132.1	1130.9	7.2920	1913.9	1130.9	6.8569	956.60	1130.9	6.7259	477.43	1130.6	6.5953	239.10	1130.4	6.4642
535.00	22148.1	1258.1	7.4606	2214.4	1258.1	7.0256	1107.4	1258.1	6.8946	553.81	1258.1	6.7639	276.72	1257.9	6.6329
650.00	25301.1	1395.8	7.6202	2530.7	1395.8	7.1852	1265.1	1395.8	7.0541	632.55	1395.8	6.9235	316.27	1395.5	6.7925
760.00	28317.1	1531.6	7.7594	2831.7	1531.6	7.3244	1415.9	1531.6	7.1933	709.80	1531.6	7.0627	353.97	1531.6	6.9317
870.00	31327.1	1670.2	7.8859	3133.3	1670.2	7.4509	1566.0	1670.2	7.3198	783.37	1670.2	7.1892	391.73	1670.2	7.0581
980.00	34336.1	1811.1	8.0040	3434.4	1811.1	7.5690	1716.8	1811.1	7.4380	858.55	1811.1	7.3074	429.20	1811.1	7.1763

T	P = 827.02kPa			P = 1102.6kPa			P = 1378.4kPa			P = 1654.0kPa			P = 2067.5kPa		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
-15.00	54.975	680.71	5.3848	40.121	676.15	5.3146	31.093	671.59	5.2548	25.152	666.75	5.2030			
10.00	61.173	702.79	5.4788	44.992	698.60	5.4127	35.284	694.42	5.3612	28.804	690.00	5.3147	22.243	683.72	5.2595
35.00	67.032	726.56	5.5588	49.406	722.94	5.4986	39.227	719.32	5.4497	32.177	715.69	5.4084	25.170	710.42	5.3546
65.00	75.289	756.19	5.6456	56.006	753.84	5.5878	44.407	751.49	5.5413	36.685	749.14	5.5020	28.892	745.62	5.4533
90.00	81.024	780.64	5.7128	60.300	779.02	5.6555	47.852	777.40	5.6101	39.554	775.74	5.5713	31.267	773.41	5.5253
150.00	95.465	837.59	5.8611	71.303	836.21	5.8042	56.813	834.82	5.7607	47.152	833.67	5.7263	37.476	831.58	5.6800
205.00	108.38	892.43	5.9820	81.068	891.51	5.9259	64.693	890.59	5.8832	53.786	889.65	5.8472	42.930	888.73	5.8049
315.00	133.97	1008.3	6.1988	100.41	1007.8	6.1431	80.267	1007.3	6.1008	66.802	1006.6	6.0657	53.363	1005.5	6.0234
425.00	159.37	1130.2	6.3876	119.45	1129.7	6.3332	95.534	1129.5	6.2913	79.272	1129.3	6.2557	63.580	1128.8	6.2138
535.00	184.53	1257.9	6.5563	138.36	1257.7	6.5018	110.68	1257.7	6.4600	92.196	1257.4	6.4252	73.785	1257.2	6.3833
650.00	210.82	1395.5	6.7158	158.13	1395.5	6.6614	126.51	1395.5	6.6195	105.44	1395.5	6.5848	84.317	1395.3	6.5429
760.00	236.04	1531.6	6.8550	177.05	1531.6	6.8006	141.65	1531.4	6.7588	118.05	1531.4	6.7240	94.453	1531.4	6.6821
870.00	261.19	1670.2	6.9815	195.90	1670.2	6.9271	156.73	1670.2	6.8852	130.66	1670.2	6.8505	104.53	1670.0	6.8086
980.00	286.17	1811.1	7.0997	214.68	1811.1	7.0453	171.81	1811.1	7.0034	143.15	1811.1	6.9686	114.54	1811.1	6.9268

T	P = 2481.1kPa			P = 3032.4kPa			P = 3583.7kPa			P = 4135.1kPa			P = 5513.4kPa		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
10.00	17.842	677.21	5.2067	13.834	668.84	5.1422	11.062	660.23	5.0861	9.0645	651.86	5.0325			
35.00	20.544	704.81	5.3065	16.255	697.57	5.2473	13.313	690.60	5.1962	11.163	683.36	5.1475	7.2510	665.24	5.0576
65.00	23.551	742.10	5.4097	18.941	737.40	5.3560	15.655	732.71	5.3099	13.222	728.01	5.2663	9.2286	716.27	5.1857
90.00	25.706	770.79	5.4851	20.695	767.56	5.4352	17.213	764.26	5.3926	14.566	761.04	5.3520	10.514	752.81	5.2768
150.00	31.032	829.73	5.6444	25.185	826.96	5.6005	21.130	824.42	5.5633	18.150	821.66	5.5287	13.318	815.18	5.4631
205.00	35.600	886.64	5.7702	28.962	884.79	5.7292	24.385	882.93	5.6949	21.023	881.07	5.6639	15.561	876.21	5.6032
315.00	44.985	1005.2	5.9890	36.279	1004.3	5.9505	30.633	1003.3	5.9183	26.505	1001.9	5.8902	19.788	999.61	5.8332
425.00	52.950	1128.3	6.1795	43.279	1127.8	6.1414	36.625	1127.2	6.1095	31.716	1126.7	6.0824	23.739	1125.3	6.0279
535.00	61.425	1256.9	6.3490	50.261	1256.5	6.3109	42.502	1256.0	6.2791	36.835	1255.8	6.2519	27.596	1254.8	6.1975
650.00	70.317	1395.3	6.5086	57.510	1395.1	6.4705	48.660	1394.9	6.4387	42.172	1394.4	6.4115	31.627	1393.9	6.3570
760.00	78.722	1531.4	6.6478	64.426	1531.2	6.6097	54.518	1530.9	6.5779	47.264	1530.7	6.5507	35.459	1530.2	6.4962
870.00	87.127	1670.0	6.7743	71.224	1669.8	6.7362	60.322	1669.8	6.7019	52.295	1669.8	6.6771	39.242	1669.3	6.6227
980.00	95.534	1810.8	6.8924	78.198	1810.8	6.8543	66.178	1810.9	6.8223	57.342	1810.6	6.7953	43.035	1810.4	6.7409

T	P = 6891.8kPa			P = 8270.1kPa			P = 9648.5kPa			P = 12405kPa			P = 11026kPa		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
65.00	6.8454	704.53	5.1132												
90.00	8.0288	744.66	5.2097												
150.00	10.421	808.48	5.4055	6.3717	737.30	5.1552	7.1234	795.31	5.3220	8.5474	847.28	5.4150	7.4461	855.60	5.4451
205.00	12.291	871.58	5.5522	10.135	866.94	5.5104	8.5983	862.07	5.4769	9.5877	987.48	5.6674	9.7078	990.05	5.6938
315.00	15.752	997.28	5.7884	13.071	994.95	5.7516	11.151	992.38	5.7214	12.452	1121.1	5.9194	12.216	1120.4	5.8913
425.00	18.982	1123.6	5.9852	15.761	1122.5	5.9492	13.452	1121.1	5.9194	14.030	1118.2	5.8683	11.821	1119.6	5.8937
535.00	22.065	1253.9	6.1556	18.369	1253.0	6.1204	15.730	1252.0	6.0911	16.216	1250.4	6.0418	13.753	1251.3	6.0654
650.00	25.308	1393.2	6.3152	21.089	1392.5	6.2800	18.076	1392.1	6.2511	14.058	1390.7	6.2030	15.814	1391.4	6.2259
760.00	28.373	1529.8	6.4544	23.654	1529.5	6.4192	20.283	1529.1	6.3903	15.782	1527.9	6.3422	17.748	1528.4	6.3610
870.00	31.414	1669.1	6.5808	26.188	1668.8	6.5457	22.460	1668.4	6.5168	17.488	1667.9	6.4686	19.664	1668.1	6.4916
980.00	34.437	1810.2	6.6990	28.722	1809.9	6.6638	24.025	1809.9	6.6350	19.187	1809.5	6.5868	21.568	1809.7	6.6102

T = °C. P = kPa

THE PROPERTIES OF SATURATED ETHYLENE

T	P	v _f	v _g	h _f	h _g	s _f	s _g	T	P	v _f	v _g	h _f	h _g	s _f	s _g
-165.00	.23		135188.	-146.32	415.54	-1.0480	4.1683	-75.00	441.31	1.9511	98.020	72.22	504.26	0.3894	2.5701
-180.00	.51		66007.	-136.03	421.43	-0.9509	3.9762	-60.00	775.02	1.9994	90.801	111.17	512.61	0.6044	2.4879
-190.00	2.08		18236.	-111.60	432.72	-0.7678	3.6544	-50.00	1048.0	2.0749	51.537	137.64	516.28	0.7074	2.4044
-140.00	6.14		6436.3	-87.35	445.12	-0.5786	3.4206	-40.00	1422.0	2.1650	37.020	164.65	518.14	0.8101	2.3262
-130.00	15.61		2697.5	-83.42	462.69	-0.4113	3.2440	-30.00	1927.8	2.2914	26.996	193.68	516.76	0.9268	2.2564
-120.00	34.44		1299.8	-38.74	458.92	-0.2225	3.0578	-20.00	2528.0	2.3849	20.142	225.46	516.67	1.0497	2.1991
-110.00	69.13		682.32	-15.69	485.10	-0.1083	2.9446	-5.00	3639.0	2.7003	12.580	286.18	503.30	1.2714	2.0811
-105.00	93.50	1.7420	518.14	-2.90	482.78	-0.0082	2.8797	5.00	4570.5	3.2157	8.3607	350.43	474.99	1.4949	1.9427
-95.00	164.28	1.7905	308.77	21.48	490.06	0.1161	2.7459	10.00	5128.9	4.4060	4.2584	400.72	397.13	1.6636	1.6508
-85.00	268.28	1.8382	199.36	46.98	497.78	0.2672	2.6527								

THE PROPERTIES OF SUPERHEATED ETHYLENE

T	P = 101.3kPa			P = 202.7kPa			P = 405.2kPa			P = 607.8kPa		
	v	h	s	v	h	s	v	h	s	v	h	s
-95.00	504.97	494.26	2.9135	260.66	501.41	2.7543	135.97	512.73	2.6324	92.551	518.75	2.5577
-85.00	536.21	507.03	2.9851	284.97	521.67	2.8532	144.65	527.65	2.6977	98.353	534.57	2.6233
-70.00	582.59	526.02	3.0854	301.03	535.30	2.9157	153.24	542.06	2.7616	104.44	550.23	2.6879
-60.00	613.38	539.24	3.1443	316.82	549.01	2.9797	161.56	556.51	2.8261	115.70	573.03	2.7816
-50.00	642.98	552.39	3.2058	332.37	562.79	3.0396	173.84	578.45	2.9136	120.56	588.59	2.8437
-40.00	673.85	565.81	3.2657	348.09	576.57	3.1035	181.88	593.59	2.9727	131.19	619.59	2.9590
-25.00	713.89	586.59	3.3496	365.56	583.95	3.1235	189.79	608.48	3.0330	145.63	668.35	3.1249
-15.00	754.00	600.70	3.478	370.88	598.52	3.1816	197.64	623.66	3.0884	158.63	710.23	3.2531
-5.00	808.47	629.69	3.5156	401.25	627.61	3.2894	205.39	639.36	3.1412	168.91	745.58	3.3529
5.00	838.18	644.97	3.5686	416.36	643.08	3.3425	213.91	671.41	3.2494	181.63	791.27	3.4816
15.00	897.64	676.08	3.6717	446.46	674.54	3.4480	229.91	713.02	3.3787	191.89	829.53	3.5756
35.00	971.38	716.98	3.8018	483.75	715.35	3.5797	255.10	748.06	3.4757	209.26	899.53	3.7514
60.00	1030.4	751.45	3.8974	513.73	750.12	3.6741	273.85	793.14	3.6033	226.82	973.40	3.9182
80.00	1104.1	796.36	4.0259	550.72	795.23	3.8081	315.01	900.93	3.8726	241.71	1040.8	4.0577
105.00	1163.1	833.99	4.1200	580.63	833.29	3.9020	341.03	974.80	4.0397	259.08	1120.5	4.2119
125.00	1266.0	903.26	4.2915	631.77	902.56	4.0779	363.22	1042.0	4.1792			
160.00	1369.0	976.63	4.4540	683.56	976.18	4.2405	389.17	1121.6	4.3333			
225.00	1457.2	1043.6	4.5896	727.96	1043.2	4.3799						
260.00	1560.1	1123.02	4.7436	779.10	1122.6	4.5343						

T	P = 1013kPa			P = 1520kPa			P = 2026kPa			P = 3039kPa		
	v	h	s	v	h	s	v	h	s	v	h	s
-50.00	54.194	519.22	2.4298	39.600	544.43	2.4293	29.335	546.57	2.3920	17.958	532.32	2.2263
-40.00	58.432	536.74	2.5079	42.551	563.02	2.5023	31.814	567.39	2.4301	19.960	558.34	2.3159
-25.00	64.263	561.80	2.6010	45.282	581.42	2.5746	34.148	587.36	2.4817	21.766	581.93	2.3988
-15.00	68.135	577.93	2.6639	47.904	599.88	2.6411	36.288	606.58	2.5845	24.809	625.44	2.5459
-5.00	71.736	594.42	2.7281	50.487	618.09	2.7020	40.157	644.44	2.7032	28.342	674.88	2.7047
5.00	75.239	611.17	2.7872	55.354	653.30	2.8131	44.761	690.47	2.8554	30.796	715.34	2.8167
15.00	78.726	627.90	2.8437	61.054	697.91	2.9601	48.196	728.66	2.9570	33.967	765.54	2.9588
35.00	85.359	661.65	2.9595	65.484	734.75	3.0569	52.409	776.68	3.1007	36.330	806.53	3.0644
60.00	93.679	704.88	3.0940	70.909	782.00	3.1971	55.726	816.63	3.1984	40.266	880.47	3.2448
80.00	99.990	740.90	3.1938	75.122	821.32	3.2956	61.367	888.60	3.3787	44.129	957.49	3.4160
105.00	107.79	787.09	3.3269	82.530	892.56	3.4709	66.936	964.17	3.5456	47.274	1027.0	3.5554
125.00	103.97	825.78	3.4248	89.802	967.40	3.6377	71.599	1032.6	3.6850	51.004	1108.6	3.7137
160.00	124.79	896.28	3.6006	95.925	1035.7	3.7733	76.974	1113.5	3.8393			
195.00	135.41	970.64	3.7674	103.01	1116.1	3.9272						
225.00	144.51	1038.5	3.9031									
260.00	155.01	1118.4	4.0570									

T	P = 4052kPa			P = 50.65kPa			P = 6078kPa			P = 8104kPa		
	v	h	s	v	h	s	v	h	s	v	h	s
5.00	12.120	516.28	2.1133	9.4991	523.84	2.0998	5.5428	444.63	1.7917	3.3873	369.09	1.6150
15.00	14.159	550.16	2.2302	12.008	573.53	2.2794	8.5626	545.78	2.1329	4.7435	459.65	1.9794
35.00	16.959	603.05	2.4061	15.170	640.93	2.4702	11.611	621.88	2.3781	7.5538	577.44	2.1855
60.00	20.164	659.07	2.5833	16.850	685.87	2.8049	13.472	671.94	2.5263	9.1955	639.48	2.3830
80.00	22.184	701.63	2.7066	19.323	742.38	2.7621	15.525	730.60	2.6826	10.896	705.61	2.5491
105.00	24.710	754.19	2.8541	20.723	786.33	2.8714	16.933	776.26	2.7995	12.077	755.07	2.6735
125.00	26.565	798.67	2.9596	23.410	863.95	3.0606	19.165	855.35	2.9977	13.984	838.37	2.8805
160.00	29.716	872.33	3.1443	25.498	932.26	3.2156	21.024	925.55	3.1490	15.450	911.10	3.0434
195.00	32.662	950.80	3.3196	27.853	1015.1	3.3836	23.067	1009.1	3.3202	17.089	997.73	3.2159
225.00	35.134	1021.2	3.4591	30.215	1098.6	3.5462	25.096	1093.7	3.4876	18.791	1084.0	3.3829
260.00	38.019	1103.5	3.6216									

T	P = 10 130kPa			P = 15 195kPa			P = 20 260kPa			P = 25 325kPa			P = 30 390kPa		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
20.00	2.8209	345.44	1.4120	2.4768	337.25	1.3345	2.3893	333.90	1.2704	2.2657	331.97	1.2333	2.1977	331.25	1.1968
35.00	3.4254	408.79	1.6241	2.7603	382.83	1.4845	2.5910	373.84	1.4029	2.4385	368.96	1.3599	2.3682	368.24	1.3209
60.00	5.1191	530.93	2.0097	3.4950	464.65	1.7459	2.9341	444.42	1.6412	2.6844	436.05	1.5742	2.5595	433.26	1.5240
80.00	6.6400	604.69	2.2162	4.0985	538.74	1.9590	3.3034	507.90	1.8213	3.0150	493.54	1.7354	2.7963	487.09	1.6712
105.00	8.2067	679.03	2.3275	5.0804	623.24	2.1982	3.8781	588.54	2.0437	3.3753	568.89	1.9385	3.0879	566.65	1.8711
125.00	9.2195	733.13	2.5594	5.8136	694.71	2.3492	4.3235	650.17	2.2023	3.6532	628.36	2.0931	3.3495	613.54	2.0142
160.00	10.803	821.63	2.7842	6.9295	782.56	2.5916	5.3064	752.09	2.4493	4.3075	736.47	2.3488	3.8081	715.58	2.2651
190.00	12.158	897.56	2.9508	7.8882	865.25	2.7773	5.9874	838.60	2.6424	4.8135	817.76	2.5420	4.2556	802.86	2.4621
225.00	13.005	986.49	3.1355	8.8834	960.22	2.9722	6.7063	937.04	2.8463	5.4590	918.73	2.7495	4.7840	904.91	2.6701
260.00	14.920	1074.4	3.3034	9.9260	1052.6	3.1485	7.4289	1033.0	3.0312	6.1804	1017.4	2.9391	5.1815	1004.7	2.8638

h = kJ/kg, v = cm³/g and s = kJ/kg·K

Datum - Sat. liquid h and s = 0 at 1 atmosphere pressure

THERMODYNAMIC DATA

T = °C. P = kPa

THE PROPERTIES OF SATURATED FREON-21

Datum - Sat. liquid h
and s = 0 at -40°C.

T	P	v _f	v _g	h _f	h _g	s _f	s _g	T	P	v _f	v _g	h _f	h _g	s _f	s _g
-40.00	9.36	0.6605	2003.3	0.00	266.42	0.0033	1.1430	20.00	152.64	0.7244	149.93	60.39	296.65	0.2310	1.0340
-35.00	12.60	0.6646	1519.0	4.95	268.94	0.2090	1.1295	25.00	183.38	0.7326	126.61	65.76	299.11	0.2469	1.0323
-30.00	16.70	0.6715	1166.5	9.80	271.43	0.0403	1.1182	30.00	215.23	0.7385	108.19	71.07	301.58	0.2654	1.0258
-25.00	21.91	0.6772	908.48	14.94	273.99	0.0529	1.1053	35.00	252.98	0.7452	92.889	76.50	304.00	0.2829	1.0213
-20.00	28.25	0.6816	715.33	19.63	276.47	0.0756	1.0957	40.00	294.99	0.7525	80.289	81.86	306.38	0.3004	1.0175
-15.00	36.13	0.6848	570.10	24.72	279.00	0.1005	1.0856	45.00	345.87	0.7595	69.598	87.45	308.74	0.3176	1.0135
-10.00	45.67	0.6905	458.76	29.71	281.54	0.1197	1.0770	50.00	395.80	0.7671	60.632	93.00	311.04	0.3351	1.0099
-5.00	57.08	0.6960	372.79	34.70	284.03	0.1385	1.0685	55.00	456.26	0.7750	53.046	98.61	313.29	0.3523	1.0065
0.00	70.75	0.7018	305.53	39.76	286.58	0.1568	1.0615	60.00	521.85	0.7828	46.552	104.33	315.49	0.3693	1.0032
5.00	86.80	0.7072	252.53	44.84	289.08	0.1755	1.0533	65.00	582.37	0.7912	40.994	110.10	317.63	0.3862	1.0001
10.00	105.65	0.7129	210.38	49.98	291.60	0.1938	1.0475	70.00	663.95	0.7998	36.223	115.95	319.74	0.4032	0.9975
15.00	127.76	0.7195	176.47	55.21	294.15	0.2116	1.0438								

THE PROPERTIES OF SUPERHEATED FREON-21

P = 8.27 kPa (Sat. Temp. = -42.00°C)				P = 13.78 kPa (Sat. Temp. = -33.44°C)				P = 27.57 kPa (Sat. Temp. = -20.50°C)				P = 68.92 kPa (Sat. Temp. = -0.61°C)			
T	v	h	s	T	v	h	s	T	v	h	s	T	v	h	s
-40.00	2268.6	266.44	1.1530	-30.00	1416.1	271.47	1.1333	-15.00	749.51	279.12	1.1080	5.00	19.97	289.37	1.0733
-35.00	2314.8	269.02	1.1640	-25.00	1447.8	274.13	1.1434	-10.00	764.60	281.81	1.1183	10.00	325.81	292.16	1.0831
-30.00	2375.6	271.60	1.1748	-15.00	1505.8	279.36	1.1645	-5.00	778.70	284.50	1.1283	15.00	331.49	295.01	1.0929
-25.00	2401.2	274.18	1.1857	-10.00	1534.7	282.02	1.1747	0.00	793.74	287.25	1.1386	20.00	338.75	297.86	1.1029
-20.00	2479.4	276.83	1.1959	-5.00	1564.4	284.72	1.1852	5.00	808.93	290.00	1.1487	25.00	343.87	300.80	1.1123
-10.00	2562.8	282.11	1.2162	0.00	1594.2	287.45	1.1942	10.00	823.42	292.79	1.1585	30.00	349.86	303.60	1.1226
-5.00	2611.1	284.81	1.2263	5.00	1623.1	290.19	1.2061	15.00	837.95	295.63	1.1684	40.00	361.84	309.53	1.1414
0.00	2660.5	287.54	1.2366	10.00	1652.5	292.98	1.2150	20.00	852.85	298.44	1.1779	45.00	367.80	312.51	1.1506
5.00	2709.8	290.28	1.2467	15.00	1682.5	295.81	1.2220	25.00	868.68	301.37	1.1880	50.00	373.79	315.50	1.1604
10.00	2758.7	293.05	1.2565	20.00	1711.4	298.62	1.2300	30.00	881.59	304.17	1.1975	55.00	379.71	318.53	1.1694
15.00	2807.6	295.85	1.2663	25.00	1740.6	301.54	1.2438	35.00	898.50	307.22	1.2063	60.00	385.68	321.58	1.1786
20.00	2856.6	298.69	1.2759	30.00	1771.7	304.42	1.2536	40.00	910.07	310.00	1.2179	65.00	391.70	324.66	1.1881
25.00	2904.6	301.57	1.2860	40.00	1829.3	310.27	1.2726	50.00	941.32	316.03	1.2353	70.00	397.65	327.78	1.1970
30.00	2955.4	304.47	1.2950	45.00	1859.1	313.22	1.2825	55.00	955.22	319.03	1.2444	75.00	403.61	330.90	1.2067
35.00	3002.0	307.41	1.3060	50.00	1887.9	316.20	1.2914	60.00	971.38	322.07	1.2535	80.00	409.56	334.09	1.2141
40.00	3052.6	310.31	1.3121	55.00	1917.4	319.23	1.3007	65.00	986.18	325.14	1.2629	85.00	415.52	337.32	1.2256
50.00	3150.4	316.27	1.3328	60.00	1947.1	322.26	1.3100	70.00	1000.3	328.26	1.2723	90.00	421.50	340.41	1.2322
55.00	3199.2	319.27	1.3421	65.00	1976.5	325.31	1.3191	75.00	1015.8	331.39	1.2804	100.00	433.39	346.98	1.2508
60.00	3248.1	322.30	1.3515	70.00	2005.4	328.43	1.3280	80.00	1031.0	334.54	1.2911	105.00	439.33	350.28	1.2595
65.00	3297.0	325.38	1.3606	75.00	2034.7	331.54	1.3377	90.00	1059.9	340.91	1.3079	110.00	445.30	353.60	1.2682
70.00	3345.9	328.49	1.3696	80.00	2065.8	334.74	1.3451	95.00	1074.0	344.16	1.3167	115.00	452.24	356.95	1.2770
75.00	3394.8	331.63	1.3786	85.00	2092.6	337.63	1.3567	100.00	1089.1	347.42	1.3254	120.00	451.10	360.33	1.2851
80.00	3443.6	334.78	1.3877	90.00	2124.8	341.03	1.3639	105.00	1104.1	350.72	1.3340	125.00	478.38	363.68	1.2948
85.00	3492.5	337.95	1.3967	100.00	2182.1	347.56	1.3819	110.00	1118.7	354.02	1.3427	130.00	484.53	367.13	1.3024
90.00	3541.4	341.14	1.4057	105.00	2211.3	350.86	1.3906	115.00	1133.3	357.35	1.3514	135.00	480.07	370.56	1.3111
100.00	3639.2	347.58	1.4205	110.00	2240.5	354.16	1.3992	120.00	1147.9	360.71	1.3600	140.00	485.27	374.04	1.3196
105.00	3688.0	350.83	1.4231	115.00	2270.3	357.49	1.4079	125.00	1162.5	364.10	1.3687	150.00	494.06	381.06	1.3362
110.00	3736.9	354.09	1.4185	120.00	2299.8	360.99	1.4153	130.00	1177.4	367.50	1.3772	155.00	496.22	384.61	1.3449
115.00	3785.8	357.38	1.4020												
120.00	3834.7	360.69	1.3675												

h = kJ/kg, v = cm³/g
and s = kJ/kg·K

T	P	v _f	v _g	h _f	h _g
15.00				4.77	384.70
40.00				45.65	412.25
65.00				89.37	446.23
90.00		0.9836	60330.	133.16	481.33
105.00		1.0106	30457.	161.47	505.11
115.00	1.40	1.0161	18935.	180.62	519.42
125.00	1.80	1.0293	11837.	201.25	539.06
140.00	3.72	1.0369	7334.3	231.96	563.62
150.00	5.26	1.0514	4909.4	253.17	583.17
160.00	7.58	1.0613	3283.7	274.42	600.00
170.00	10.57	1.0680	2482.7	296.04	617.60
180.00	14.38	1.0732	1835.3	315.35	633.63
195.00	21.76	1.0932	1176.9	353.02	668.50
205.00	28.45	1.0987	910.04	378.14	689.78
215.00	36.04	1.1096	724.66	403.26	711.21
225.00	48.99	1.1256	544.14	428.28	727.36
240.00	66.05	1.1345	389.01	465.79	759.09
250.00	84.35	1.1417	311.37	491.48	780.56
260.00	103.38	1.1549	254.71	516.28	802.33
270.00	129.06	1.1750	209.57	540.76	820.25
280.00	154.82	1.1777	155.57	569.16	843.31
295.00	211.24	1.2058	119.63	609.53	874.94
305.00	251.31	1.2183	103.95	638.84	899.45
315.00	295.05	1.2356	92.530	668.14	924.62
325.00	338.49	1.2474	78.184	697.44	947.01
340.00	441.52	1.2720	59.728	741.40	985.10
350.00	519.32	1.2956	50.548	770.70	1007.0
360.00	606.48	1.3172	43.325	800.00	1030.2
370.00	699.05	1.3281	38.965	829.30	1055.5
380.00	802.10	1.3522	34.196	858.60	1077.3
400.00	1047.9	1.3986	26.686	917.21	1126.0

THE PROPERTIES OF SATURATED DOWTHERM A

Datum - Sat. liquid h = 0 at 12°C.

Properties of Saturated Dichloromonofluoromethane (F-21) *

Temp., °F. <i>t</i>	Abs. pressure, lb./sq. in. <i>p</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./lb. (°R.)		Temp., °F. <i>t</i>	Abs. pressure, lb./sq. in. <i>p</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./lb. (°R.)	
		Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>			Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>
-40	1.358	0.01058	32.09	0.00	114.56	0.0000	0.2730	70	23.08	0.1164	2.300	26.49	127.79	.0559	.2471
-30	1.888	.01066	23.61	2.36	115.76	.0055	.2695	80	27.96	0.1176	1.923	29.03	128.98	.0606	.2458
-20	2.578	.01075	17.66	4.71	116.96	.0109	.2663	86	31.23	.01183	1.733	30.56	129.68	.0634	.2450
-10	3.463	.01084	13.43	7.07	118.17	.0162	.2633	90	33.58	.01188	1.619	31.59	130.14	.0652	.2446
0	4.582	.01093	10.35	9.44	119.37	.0214	.2606	100	40.04	.01200	1.371	34.18	131.29	.0699	.2434
5	5.243	.01097	9.132	10.63	119.97	.0240	.2593	110	47.40	.01213	1.169	36.79	132.42	.0745	.2424
10	5.978	.01102	8.085	11.81	120.57	.0265	.2581	120	55.75	.01226	1.001	39.46	133.53	.0791	.2414
20	7.699	.01112	6.392	14.21	121.78	.0316	.2559	130	65.15	.01240	0.8623	42.13	134.61	.0837	.2405
30	9.793	.01122	5.112	16.61	122.98	.0365	.2538	140	75.72	.01254	.7457	44.86	135.66	.0882	.2396
40	12.32	.01132	4.130	19.04	124.19	.0414	.2519	150	87.51	.01269	.6476	47.62	136.68	.0927	.2388
50	15.33	.01142	3.370	21.49	125.39	.0463	.2502	160	100.6	.01284	.5646	50.43	137.69	.0972	.2381
60	18.90	.01153	2.773	23.98	126.60	.0511	.2486								

* Courtesy Kinetic Chemicals, Inc.

Properties of Superheated Dichloromonofluoromethane (F-21) *

v, volume, cu. ft./lb.; *h*, enthalpy, B.t.u./lb.; *s*, entropy, B.t.u./lb. (°R.)
Parenthetic figures after pressures are saturation temperatures

Temp., °F. <i>t</i>	Abs. pressure 1.2 lb./sq. in. (-43.6°F.)			Temp., °F. <i>t</i>	Abs. pressure 2 lb./sq. in. (-28.2°F.)			Temp., °F. <i>t</i>	Abs. pressure 4 lb./sq. in. (-4.9°F.)			Temp., °F. <i>t</i>	Abs. pressure 10 lb./sq. in. (30.9°F.)		
	<i>v</i>	<i>h</i>	<i>s</i>		<i>v</i>	<i>h</i>	<i>s</i>		<i>v</i>	<i>h</i>	<i>s</i>		<i>v</i>	<i>h</i>	<i>s</i>
Sat.	36.02	114.13	0.2744	Sat.	22.37	115.98	0.2689	Sat.	11.74	118.78	0.2619	Sat.	5.014	123.10	0.2536
-40	36.34	114.57	.2754	-20	22.80	116.99	.2712	0	11.87	119.40	.2633	40	5.112	124.30	.2561
-30	37.21	115.80	.2783	-10	23.33	118.24	.2740	10	12.14	120.66	.2660	50	5.219	125.63	.2587
-20	38.08	117.03	.2812	0	23.85	119.49	.2768	20	12.40	121.95	.2687	60	5.326	126.99	.2613
-10	38.95	118.28	.2840	10	24.38	120.76	.2795	30	12.66	123.25	.2714	70	5.443	128.36	.2639
0	39.83	119.54	.2867	20	24.90	122.04	.2822	40	12.93	124.57	.2741	80	5.540	129.74	.2665
10	40.70	120.80	.2894	30	25.43	123.34	.2849	50	13.19	125.90	.2767	90	5.646	131.13	.2691
20	41.57	122.08	.2921	40	25.95	124.65	.2876	60	13.45	127.25	.2793	100	5.753	132.54	.2716
30	42.44	123.38	.2948	50	26.47	125.98	.2902	70	13.72	128.61	.2819	110	5.860	133.95	.2741
40	43.32	124.69	.2975	60	27.00	127.33	.2928	80	13.98	129.99	.2845	120	5.966	135.38	.2766
50	44.19	126.01	.3001	70	27.52	128.69	.2954	90	14.24	131.37	.2870	130	6.072	136.82	.2791
60	45.06	127.35	.3027	80	28.05	130.07	.2979	100	14.51	132.77	.2895	140	6.178	138.28	.2815
70	45.93	128.71	.3053	90	28.57	131.45	.3005	110	14.77	134.18	.2920	150	6.285	139.75	.2840
80	46.80	130.09	.3079	100	29.09	132.85	.3030	120	15.03	135.61	.2945	160	6.391	141.24	.2864
90	47.67	131.48	.3104	110	29.62	134.26	.3055	130	15.29	137.04	.2970	170	6.497	142.75	.2888
100	48.54	132.88	.3129	120	30.14	135.68	.3080	140	15.56	138.49	.2994	180	6.603	144.27	.2912
110	49.42	134.29	.3154	130	30.66	137.12	.3104	150	15.82	139.96	.3019	190	6.709	145.79	.2936
120	50.29	135.71	.3179	140	31.19	138.57	.3129	160	16.08	141.45	.3043	200	6.815	147.33	.2959
130	51.16	137.14	.3203	150	31.71	140.03	.3153	170	16.35	142.95	.3067	210	6.921	148.89	.2983
140	52.03	138.59	.3228	160	32.23	141.52	.3177	180	16.61	144.46	.3091	220	7.027	150.46	.3006
150	52.90	140.06	.3252	170	32.76	143.02	.3201	190	16.87	145.98	.3114	230	7.133	152.05	.3029
160	53.77	141.55	.3276	180	33.28	144.53	.3225	200	17.13	147.52	.3138	240	7.239	153.65	.3052
170	54.64	143.05	.3300	190	33.80	146.05	.3249	210	17.39	149.08	.3161	250	7.344	155.26	.3075
180	55.51	144.56	.3324	200	34.33	147.59	.3272	220	17.66	150.65	.3184	260	7.450	156.88	.3098
190	56.38	146.08	.3348	210	34.85	149.14	.3296	230	17.92	152.23	.3207	270	7.555	158.52	.3120
200	57.25	147.61	.3371	220	35.37	150.71	.3319	240	18.18	153.82	.3230	280	7.661	160.17	.3143
210	58.12	149.16	.3394	230	35.89	152.29	.3342	250	18.44	155.43	.3253	290	7.767	161.84	.3165
220	58.99	150.73	.3418	240	36.42	153.88	.3365	260	18.70	157.05	.3276	300	7.872	163.52	.3187
230	59.86	152.31	.3441	250	36.94	155.59	.3388	270	18.97	158.68	.3298	310	7.978	165.21	.3210
240	60.73	153.90	.3464												
250	61.60	155.51	.3486												

* Courtesy of Kinetic Chemicals, Inc.

Properties of Saturated Propane*

Temp., °F.	Abs. pressure, lb./sq. in. <i>p</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./lb. (°R.)		Temp., °F.	Abs. pressure, lb./sq. in. <i>p</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./lb. (°R.)	
		Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>			Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>
-80	5.65	0.0265	16.2	162.6	354.0	0.8794	1.3832	70	124.3	0.03209	.854	245.7	394.4	1.0624	1.3427
-70	7.48	0.0268	12.5	167.6	357.0	.8927	1.3781	80	143.6	0.03269	.745	251.9	396.4	1.0737	1.3413
-60	9.78	0.02703	9.77	172.7	360.0	.9060	1.3740	90	165.0	0.03329	.643	258.2	398.3	1.0850	1.3400
-50	12.60	0.02733	7.73	177.8	362.8	.9188	1.3702	100	188.7	0.03390	.558	264.6	400.2	1.0963	1.3388
-40	16.00	0.02763	6.16	183.0	365.7	.9315	1.3670	110	214.8	0.03452	.487	271.1	401.9	1.1080	1.3378
-30	20.18	0.02794	5.02	188.4	368.6	.9441	1.3640	120	243.4	0.03532	.426	278.0	403.8	1.1195	1.3368
-20	25.05	0.02826	4.06	193.8	371.5	.9568	1.3610	130	274.5	0.03612	.370	285.2	405.4	1.1310	1.3356
-10	30.95	0.02859	3.33	199.4	374.4	.9690	1.3582	140	308.4	0.03702	.320	292.7	407.0	1.1430	1.3347
0	37.81	0.02893	2.74	205.0	377.2	.9812	1.3555	150	345.4	0.03817	.278	300.2	408.2	1.1552	1.3336
10	45.85	0.02930	2.30	210.7	380.0	.9932	1.3531	160	385.0	0.03962	.240	308.4	408.8	1.1680	1.3303
20	55.00	0.02970	1.93	216.6	382.6	1.0050	1.3510	170	426.0	0.04132	.208	317.5	408.6	1.1816	1.3272
30	65.70	0.03011	1.60	222.3	385.1	1.0167	1.3491	180	473.2	0.04367	.180	327.5	407.6	1.1970	1.3223
40	77.80	0.03055	1.33	227.9	387.5	1.0283	1.3473	190	523.4	0.04712	.149	339.2	404.6	1.2140	1.3156
50	91.50	0.03101	1.14	233.8	389.9	1.0398	1.3456	200	575.0	0.0521	.113	353.5	398.3	1.2360	1.3040
60	106.9	0.03150	0.984	239.6	392.2	0.0511	1.3441								

* Stearns and George, *Ind. Eng. Chem.*, 35, 602 (1943); with permission.

Properties of Superheated Propane*

v, volume, cu. ft./lb.; h, enthalpy, B.t.u./lb.; s, entropy, B.t.u./lb. (°R.); p, absolute pressure, lb./sq. in.
Parenthetic figures after pressures are saturation temperatures

Temp., °F.	7.35 (-70.7°F.)			12.24 (-51°F.)			14.696 (-43.708°F.)			20 (-30.30°F.)		
	v	h	s	v	h	s	v	h	s	v	h	s
Sat.	12.72	356.5	1.3785	7.90	362.75	1.3718	6.66	364.6	1.3681	5.050	368.4	1.3640
-60	13.11	360.3	1.3869									
-40	13.79	366.8	1.4023	8.182	366.2	1.3796	6.775	365.8	1.3710			
-20	14.47	373.4	1.4177	8.591	373.0	1.3948	7.123	372.8	1.3866	5.186	372.2	1.3719
0	15.14	380.2	1.4329	9.000	379.9	1.4100	7.471	379.8	1.4020	5.439	379.3	1.3873
20	15.82	387.4	1.4481	9.409	387.0	1.4252	7.816	386.8	1.4272	5.695	386.5	1.4025
40	16.50	394.8	1.4633	9.818	394.4	1.4404	8.160	394.3	1.4324	5.951	393.9	1.4177
60	17.17	402.4	1.4785	10.23	402.0	1.4556	8.503	401.9	1.4474	6.207	401.6	1.4327
80	17.85	410.2	1.4937	10.64	410.0	1.4706	8.844	409.8	1.4623	6.461	409.4	1.4477
100	18.52	418.4	1.5088	11.050	418.1	1.4854	9.187	418.0	1.4772	6.716	417.6	1.4625
120	19.20	426.6	1.5235	11.45	426.2	1.5002	9.528	426.2	1.4918	6.969	425.9	1.4771
140	19.88	435.2	1.5380	11.86	435.9	1.5148	9.869	434.8	1.5064	7.221	434.6	1.4917
160	20.55	444.1	1.5524	12.27	443.8	1.5293	10.21	443.7	1.5210	7.473	443.5	1.5063
180	21.23	453.1	1.5668	12.68	452.9	1.5437	10.51	452.9	1.5256	7.725	452.6	1.5209
200	21.90	462.5	1.5812	13.09	462.4	1.5581	10.84	462.4	1.5500	7.977	461.9	1.5355
Temp., °F.	30 (-11.52°F.)			40 (+2.90°F.)			60 (24.80°F.)			80 (41.69°F.)		
	v	h	s	v	h	s	v	h	s	v	h	s
Sat.	3.30	374.00	1.3588	2.61	378.2	1.3548	1.76	383.8	1.3501	1.32	387.8	1.3470
0	3.559	378.3	1.3678									
20	3.735	385.7	1.3830	2.753	384.7	1.3684						
40	3.911	393.1	1.3980	2.889	392.2	1.3838	1.863	390.1	1.3630			
60	4.087	400.8	1.4130	3.025	400.0	1.3990	1.959	398.2	1.3789	1.424	396.1	1.3624
80	4.261	408.7	1.4280	3.159	408.0	1.4140	2.053	406.3	1.3948	1.500	404.6	1.3785
100	4.432	417.0	1.4428	3.289	416.2	1.4290	2.145	414.8	1.4102	1.573	413.2	1.3940
120	4.602	425.4	1.4576	3.419	424.6	1.4440	2.235	423.4	1.4254	1.644	421.8	1.4094
140	4.772	434.0	1.4724	3.549	433.4	1.4588	2.325	432.2	1.4404	1.714	430.9	1.4248
160	4.942	443.0	1.4872	3.679	442.5	1.4736	2.415	441.4	1.4554	1.784	440.3	1.4402
180	5.112	452.2	1.5020	3.809	451.8	1.4884	2.505	450.8	1.4703	1.852	449.8	1.4554
200	5.282	461.8	1.5168	3.939	461.3	1.5030	2.593	460.5	1.4851	1.920	459.7	1.4704
Temp., °F.	100 (55.62°F.)			130 (73.20°F.)			160 (87.71°F.)					
	v	h	s	v	h	s	v	h	s	v	h	s
Sat.	1.06	391.2	1.3448	0.8165	395.0	1.3424	0.6685	397.9				
60	1.094	393.5	1.3488									
80	1.164	402.6	1.3656	.8456	398.8	1.3486						
100	1.227	411.4	1.3816	.9045	408.3	1.3659	.6969	404.7				
120	1.289	420.3	1.3962	.9588	417.8	1.3822	.7464	414.6				
140	1.347	429.6	1.4130	1.006	427.3	1.3987	.7908	424.8				
160	1.400	439.1	1.4286	1.052	437.2	1.4150	.8319	434.9				
180	1.460	448.8	1.4440	1.098	447.0	1.4310	.8712	445.1				
200	1.516	458.8	1.4598	1.143	457.1	1.4468	.9093	455.4				
Temp., °F.	190 (100.50°F.)			220 (111.85°F.)			250 (122.12°F.)			300 (137.55°F.)		
	v	h	s	v	h	s	v	h	s	v	h	s
Sat.	0.5540	400.4	1.3388	0.4738	402.5	1.3375	0.4130	404.2	1.3355	0.3332	406.6	1.3345
120	.5995	411.6	1.3580	.4911	407.7	1.3460						
140	.6415	422.1	1.3759	.5314	419.2	1.3650	.4473	415.9	1.3550	.3392	408.7	1.3372
160	.6792	432.4	1.3930	.5673	429.8	1.3827	.4816	427.1	1.3732	.3745	422.0	1.3580
180	.7144	443.0	1.4096	.5998	440.6	1.3999	.5121	438.0	1.3908	.4037	433.2	1.3765
200	.7472	453.4	1.4255	.6302	451.2	1.4161	.5408	448.9	1.4074	.4303	444.9	1.3944

* Stearns and George, *Ind. Eng. Chem.*, 25, 602 (1943); with permission.

Properties of Saturated Dowtherm A*
(73.5 % diphenyloxide, 26.5 % diphenyl)

Temp., °F. <i>t</i>	Abs. pressure, lb./sq. in. <i>p</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.	
		Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>
53.6				0	164
100				18.0	176
150				38.4	192
200		0.0160	833	60.0	210
220		.0162	500	69.0	217
240	0.20	.0163	294	78.2	224
260	.29	.0165	179	87.7	232
280	.49	.0166	125	97.5	240
300	.74	.0168	83.3	108.0	250
320	1.1	.0170	52.6	118.0	258
340	1.6	.0171	38.5	128.0	266
360	2.2	.0173	27.8	138.0	275
380	3.0	.0175	20.0	150.0	286
400	4.1	.0176	14.7	162.0	296
420	5.4	.0178	11.4	174.0	306
440	6.9	.0180	8.40	186.0	316
460	8.8	.0181	6.54	198.0	326
480	12	.0183	5.13	210.0	334
500	15	.0185	4.08	222.0	345
520	19	.0188	3.23	234.0	354
540	24	.0190	2.44	247.0	365
560	30	.0193	2.00	260.0	375
580	36	.0195	1.67	274.0	386
600	43	.0198	1.47	288.0	398
620	51	.0201	1.20	302.0	409
640	62	.0204	0.980	316.0	421
660	74	.0207	.826	330.0	432
680	88	.0211	.694	344.0	443
700	103	.0213	.617	358.0	455
720	120	.0218	.526	372.0	465
750	150	.0225	.417	393.0	482

* Dow Chemical Co.

Properties of Saturated Ammonia*

Temp., °F. <i>t</i>	Abs. pressure, lb./sq. in. <i>p</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./lb. (°R.)		Temp., °F. <i>t</i>	Abs. pressure, lb./sq. in. <i>p</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./lb. (°R.)	
		Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>			Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>
-60	5.55	0.02278	44.73	-21.2	589.6	-0.0517	1.4769	24	52.59	5.443	69.1	618.9	.1528	1.2897
-50	7.67	.02299	33.08	-10.6	593.7	-.0256	1.4497	28	57.28	5.021	73.5	619.9	.1618	1.2825
-40	10.41	.02322	24.86	0.0	597.6	.0000	1.4242	32	62.29	4.637	77.9	621.0	.1708	1.2755
-30	13.90	18.97	10.7	601.4	.0250	1.4001	36	67.63	4.289	82.3	622.0	.1797	1.2686
-20	18.30	.02369	14.68	21.4	605.0	.0497	1.3774	40	73.32	.02533	3.971	86.8	623.0	.1885	1.2618
-16	20.34	13.29	25.6	606.4	.0594	1.3686	50	89.19	.02564	3.294	97.9	625.2	.2105	1.2453
-12	22.56	12.06	30.0	607.8	.0690	1.3600	60	107.6	.02597	2.751	109.2	627.3	.2322	1.2294
-8	24.97	10.97	34.3	609.2	.0786	1.3516	70	128.8	.02632	2.312	120.5	629.1	.2537	1.2140
-4	27.59	9.991	38.6	610.5	.0880	1.3433	80	153.0	.02668	1.955	132.0	630.7	.2749	1.1991
0	30.42	.02419	9.116	42.9	611.8	.0975	1.3352	90	180.6	.02707	1.661	143.5	632.0	.2958	1.1846
4	33.47	8.333	47.2	613.0	.1069	1.3273	100	211.9	.02747	1.419	155.2	633.0	.3166	1.1705
8	36.77	7.629	51.6	614.3	.1162	1.3195	110	247.0	.02790	1.217	167.0	633.7	.3372	1.1566
12	40.31	6.996	56.0	615.5	.1254	1.3118	120	286.4	.02836	1.047	179.0	634.0	.3576	1.1427
16	44.12	6.425	60.3	616.6	.1346	1.3043	125	307.8	.02860	0.973	183.9	634.0	.3659	1.1372
20	48.21	.02474	5.910	64.7	617.8	.1437	1.2969								

* U. S. Bur. Standards Circ. 142, 1923.

Table 204. Properties of Superheated Ammonia*

v, volume, cu. ft./lb.; *h*, enthalpy, B.t.u./lb.; *s*, entropy, B.t.u./lb. (°R.)
Absolute pressure, lb. per sq. in. (saturation temperature, °F., in parentheses)

Temp., °F.	5 (-63.11)			7 (-52.88°)			10 (-41.34)			14 (-29.76)			18 (-20.61)		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
Sat.	49.31	588.5	1.4857	36.01	592.5	1.4574	25.81	597.1	1.4276	18.85	601.4	1.3996	14.90	604.8	1.3787
-50	51.05	595.2	1.5025	36.29	594.0	1.4611									
-40	52.36	600.3	1.5149	37.25	599.3	1.4739									
-30	53.67	605.4	1.5269	38.19	604.5	1.4861	26.58	603.2	1.4420						
-20	54.97	610.4	1.5385	39.13	609.6	1.4979	27.26	608.5	1.4542	19.33	606.8	1.4119	14.93	605.1	1.3795
-10	56.26	615.4	1.5498	40.07	614.7	1.5094	27.92	613.7	1.4659	19.82	612.2	1.4241	15.32	610.7	1.3921
0	57.55	620.4	1.5608	41.00	619.8	1.5206	28.58	618.9	1.4773	20.30	617.6	1.4358	15.70	616.2	1.4042
10	58.84	625.4	1.5716	41.93	624.9	1.5314	29.24	624.0	1.4884	20.78	622.8	1.4472	16.08	621.6	1.4158
20	60.12	630.4	1.5821	42.85	629.9	1.5421	29.90	629.1	1.4992	21.26	628.0	1.4582	16.46	626.9	1.4270
30	61.41	635.4	1.5925	43.77	635.0	1.5525	30.55	634.2	1.5097	21.73	633.2	1.4688	16.83	632.2	1.4380
40	62.69	640.4	1.6026	44.69	640.0	1.5627	31.20	639.3	1.5200	22.20	638.4	1.4793	17.20	637.5	1.4486
50	63.96	645.5	1.6125	45.61	645.0	1.5727	31.85	644.4	1.5301	22.67	643.6	1.4896	17.57	642.7	1.4590
60	65.24	650.5	1.6223	46.53	650.1	1.5825	32.49	649.5	1.5400	23.14	648.7	1.4996	17.94	647.9	1.4691
70	66.51	655.5	1.6319	47.44	655.2	1.5921	33.14	654.6	1.5497	23.60	653.9	1.5094	18.30	653.1	1.4790
80	67.79	660.6	1.6413	48.36	660.2	1.6016	33.78	659.7	1.5593	24.06	659.0	1.5191	18.67	658.4	1.4887

* U. S. Bur. Standards Circ. 142, 1923.

Properties of Superheated Ammonia*—(Continued)

Temp., °F.	5 (−63.11)			7 (−52.88°)			10 (−41.34)			14 (−29.76)			18 (−20.61)		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
90	69.06	665.6	1.6506	49.27	665.3	1.6110	34.42	664.8	1.5687	24.53	664.2	1.5285	19.03	663.6	1.4983
100	70.33	670.7	1.6598	50.18	670.4	1.6202	35.07	670.0	1.5779	24.99	669.4	1.5378	19.39	668.8	1.5077
110	71.60	675.8	1.6689	51.09	675.5	1.6292	35.71	675.1	1.5870	25.45	674.5	1.5470	19.75	674.0	1.5169
120	72.87	680.9	1.6778	52.00	680.7	1.6382	36.35	680.3	1.5960	25.91	679.7	1.5560	20.11	679.2	1.5260
130	74.14	686.1	1.6865	52.91	685.8	1.6470	36.99	685.4	1.6049	26.37	684.9	1.5649	20.47	684.4	1.5349
140	75.41	691.2	1.6952	53.82	691.0	1.6557	37.62	690.6	1.6136	26.83	690.1	1.5737	20.83	689.7	1.5438
150	76.68	696.4	1.7038	54.73	696.2	1.6643	38.26	695.8	1.6222	27.29	695.4	1.5824	21.19	694.9	1.5525
160	77.95	701.6	1.7122	55.63	701.4	1.6727	38.90	701.1	1.6307	27.74	700.6	1.5909	21.54	700.2	1.5610
170	79.21	706.8	1.7206	56.54	706.6	1.6811	39.54	706.3	1.6391	28.20	705.9	1.5993	21.90	705.5	1.5695
180	80.48	712.1	1.7289	57.45	711.9	1.6894	40.17	711.6	1.6474	28.66	711.2	1.6076	22.26	710.8	1.5778
190	40.81	716.9	1.6556	29.11	716.5	1.6159	22.61	716.1	1.5861
200	41.45	722.2	1.6637	29.57	721.8	1.6240	22.97	721.4	1.5943
220	23.68	732.2	1.6103

Temp., °F.	24 (−9.58)			30 (−0.57)			38 (9.42)			48 (19.80)		
	v	h	s	v	h	s	v	h	s	v	h	s
Sat.	11.39	608.6	1.3549	9.236	611.6	1.3364	7.396	614.7	1.3168	5.934	617.7	1.2973
0	11.67	614.1	1.3670	9.250	611.9	1.3371
10	11.96	619.7	1.3791	9.492	617.8	1.3497	7.407	615.0	1.3175
20	12.25	625.2	1.3907	9.731	623.5	1.3618	7.603	621.0	1.3301	5.937	617.8	1.2976
30	12.54	630.7	1.4019	9.966	629.1	1.3733	7.795	626.9	1.3422	6.096	624.0	1.3103
40	12.82	636.1	1.4128	10.20	634.6	1.3845	7.983	632.6	1.3538	6.251	630.0	1.3225
50	13.11	641.4	1.4234	10.43	640.1	1.3953	8.170	638.3	1.3650	6.404	635.9	1.3341
60	13.39	646.7	1.4337	10.65	645.5	1.4059	8.353	643.8	1.3758	6.554	641.6	1.3453
70	13.66	652.0	1.4438	10.88	650.9	1.4161	8.535	649.3	1.3863	6.702	647.3	1.3561
80	13.94	657.3	1.4537	11.10	656.2	1.4261	8.716	654.8	1.3965	6.848	652.9	1.3666
90	14.22	662.6	1.4634	11.33	661.6	1.4359	8.895	660.2	1.4065	6.993	658.5	1.3768
100	14.49	667.8	1.4729	11.55	666.9	1.4456	9.073	665.6	1.4163	7.137	664.0	1.3868
110	14.76	673.1	1.4822	11.77	672.2	1.4550	9.250	671.0	1.4258	7.280	669.5	1.3965
120	15.04	678.4	1.4914	11.99	677.5	1.4642	9.426	676.4	1.4352	7.421	675.0	1.4061
130	15.31	683.6	1.5004	12.21	682.9	1.4733	9.602	681.8	1.4444	7.562	680.5	1.4154
140	15.58	688.9	1.5093	12.43	688.2	1.4823	9.776	687.2	1.4534	7.702	685.9	1.4246
150	15.85	694.2	1.5180	12.65	693.5	1.4911	9.950	692.6	1.4623	7.842	691.4	1.4336
160	16.12	699.5	1.5266	12.87	698.8	1.4998	10.12	698.0	1.4711	7.981	696.8	1.4425
170	16.39	704.8	1.5352	13.08	704.2	1.5083	10.30	703.3	1.4797	8.119	702.3	1.4512
180	16.66	710.2	1.5436	13.30	709.6	1.5168	10.47	708.7	1.4883	8.257	707.7	1.4598
190	16.93	715.5	1.5518	13.52	714.9	1.5251	10.64	714.2	1.4966	8.395	713.2	1.4683
200	17.20	720.9	1.5600	13.73	720.3	1.5334	10.81	719.6	1.5049	8.532	718.7	1.4766
220	17.73	731.7	1.5761	14.16	731.1	1.5495	11.16	730.5	1.5212	8.805	729.6	1.4930
240	18.27	742.6	1.5919	14.59	742.0	1.5653	11.50	741.4	1.5371	9.077	740.6	1.5090
260	15.02	753.0	1.5808	11.84	752.4	1.5526	9.348	751.7	1.5246
280	12.18	763.5	1.5678	9.619	762.9	1.5399
300	9.888	774.1	1.5548

Temp., °F.	60 (30.21)			80 (44.40)			100 (56.05)			120 (66.02)		
	v	h	s	v	h	s	v	h	s	v	h	s
Sat.	4.805	620.5	1.2787	3.655	624.0	1.2545	2.952	626.5	1.2356	2.476	628.4	1.2201
30	4.933	626.8	1.2913	3.712	627.7	1.2619
40	5.060	632.9	1.3035	3.812	634.3	1.2745	2.985	629.3	1.2409
50	5.184	639.0	1.3152
60	5.307	644.9	1.3265	3.909	640.6	1.2866	3.068	636.0	1.2539	2.505	631.3	1.2255
70	5.428	650.7	1.3373	4.005	646.7	1.2981	3.149	642.6	1.2661	2.576	638.3	1.2386
80	5.547	656.4	1.3479	4.098	652.8	1.3092	3.227	649.0	1.2778	2.645	645.0	1.2510
90	5.665	662.1	1.3581	4.190	658.7	1.3199	3.304	655.2	1.2891	2.712	651.6	1.2628
100	5.781	667.7	1.3681	4.281	664.6	1.3303	3.380	661.3	1.2999	2.778	658.0	1.2741
110
120	5.897	673.3	1.3778	4.371	670.4	1.3404	3.454	667.3	1.3104	2.842	664.2	1.2850
130	6.012	678.9	1.3873	4.460	676.1	1.3502	3.527	673.3	1.3206	2.905	670.4	1.2956
140	6.126	684.4	1.3966	4.548	681.8	1.3598	3.600	679.2	1.3305	2.967	676.5	1.3058
150	6.239	689.9	1.4058	4.635	687.5	1.3692	3.672	685.0	1.3401	3.029	682.5	1.3157
160	6.352	695.5	1.4148	4.722	693.2	1.3784	3.743	690.8	1.3495	3.089	688.4	1.3254
170	6.464	701.0	1.4236	4.808	698.8	1.3874	3.813	696.6	1.3588	3.149	694.3	1.3348
180	6.576	706.5	1.4323	4.893	704.4	1.3963	3.883	702.3	1.3678	3.209	700.2	1.3441
190	6.687	712.0	1.4409	4.978	710.0	1.4050	3.952	708.0	1.3767	3.268	706.0	1.3531
200	6.798	717.5	1.4493	5.063	715.6	1.4136	4.021	713.7	1.3854	3.326	711.8	1.3620
210	6.909	723.1	1.4576	5.147	721.3	1.4220	4.090	719.4	1.3940	3.385	717.6	1.3707
220	7.019	728.6	1.4658	5.231	726.9	1.4304	4.158	725.1	1.4024	3.442	723.1	1.3793
230	7.128	734.1	1.4739	5.315	732.5	1.4386	4.226	730.8	1.4108	3.500	729.2	1.3877
240	7.238	739.7	1.4819	5.398	738.1	1.4467	4.294	736.5	1.4190	3.557	734.9	1.3960
250	7.347	745.2	1.4896	5.482	743.8	1.4547	4.361	742.2	1.4271	3.614	740.7	1.4042
260	5.565	749.4	1.4626	4.428	747.9	1.4350	3.671	746.5	1.4123
270	3.727	752.2	1.4202
280	7.675	762.1	1.5130	5.730	760.7	1.4781	4.562	759.4	1.4507	3.783	758.0	1.4281
290	7.782	767.3	1.5201	5.844	772.1	1.4933	4.695	770.8	1.4660	3.839	763.8	1.4359
300	3.895	769.6	1.4435

Properties of Superheated Ammonia* (Concluded)

Temp., °F.	140 (74.79)			160 (82.64)			180 (89.78)		
	v	h	s	v	h	s	v	h	s
Sat.	2.132	629.9	1.2068	1.872	631.1	1.1952	1.667	632.0	1.1850
80	2.166	633.8	1.2140						
90	2.228	640.9	1.2272	1.914	636.6	1.2055	1.668	632.2	1.1853
100	2.288	647.8	1.2396	1.969	643.9	1.2186	1.720	639.9	1.1992
110	2.347	654.5	1.2515	2.023	651.0	1.2311	1.770	647.3	1.2123
120	2.404	661.1	1.2628	2.075	657.8	1.2429	1.818	654.4	1.2247
130	2.460	667.4	1.2738	2.125	664.4	1.2542	1.865	661.3	1.2364
140	2.515	673.7	1.2843	2.175	670.9	1.2652	1.910	668.0	1.2477
150	2.569	679.9	1.2945	2.224	677.2	1.2757	1.955	674.6	1.2586
160	2.622	686.0	1.3045	2.272	683.5	1.2859	1.999	681.0	1.2691
170	2.675	692.0	1.3141	2.319	689.7	1.2958	2.042	687.3	1.2792
180	2.727	698.0	1.3236	2.365	695.8	1.3054	2.084	693.6	1.2891
190	2.779	704.0	1.3328	2.411	701.9	1.3148	2.126	699.8	1.2987
200	2.830	709.9	1.3418	2.457	707.9	1.3240	2.167	705.9	1.3081
210	2.880	715.8	1.3507	2.502	713.9	1.3331	2.208	712.0	1.3172
220	2.931	721.6	1.3594	2.547	719.9	1.3419	2.248	718.1	1.3262
230	2.981	727.5	1.3679	2.591	725.8	1.3506	2.288	724.1	1.3350
240	3.030	733.3	1.3763	2.635	731.7	1.3591	2.328	730.1	1.3436
250	3.080	739.2	1.3846	2.679	737.6	1.3675	2.367	736.1	1.3521
260	3.129	745.0	1.3928	2.723	743.5	1.3757	2.407	732.0	1.3605
270	3.179	750.8	1.4008	2.766	749.4	1.3838	2.446	748.0	1.3687
280	3.227	756.7	1.4088	2.809	755.3	1.3919	2.484	753.9	1.3768
290	3.275	762.5	1.4166	2.852	761.2	1.3998	2.523	759.9	1.3847
300	3.323	768.3	1.4243	2.895	767.1	1.4076	2.561	765.8	1.3926
320	3.420	780.0	1.4395	2.980	778.9	1.4229	2.637	777.7	1.4081
340				3.064	790.7	1.4379	2.713	789.6	1.4231

Temp., °F.	200 (96.34)			220 (102.42)			240 (108.09)			260 (113.42)			300 (123.21)		
	v	h	s	v	h	s	v	h	s	v	h	s	v	h	s
Sat.	1.502	632.7	1.1756	1.367	633.2	1.1671	1.253	633.6	1.1592	1.155	633.9	1.1518	0.999	634.0	1.1383
110	1.567	643.4	1.1947	1.400	639.4	1.1781	1.261	635.3	1.1621						
120	1.612	650.9	1.2077	1.443	647.3	1.1917	1.302	643.5	1.1764	1.182	639.5	1.1617			
130	1.656	658.1	1.2200	1.485	654.8	1.2045	1.342	651.3	1.1898	1.220	647.8	1.1757	1.023	640.1	1.1487
140	1.698	665.0	1.2317	1.525	662.0	1.2167	1.380	658.8	1.2025	1.257	655.6	1.1889	1.058	648.7	1.1632
150	1.740	671.8	1.2429	1.564	669.0	1.2281	1.416	666.1	1.2145	1.292	663.1	1.2014	1.091	656.9	1.1767
160	1.780	678.4	1.2537	1.601	675.8	1.2394	1.452	673.1	1.2259	1.326	670.4	1.2132	1.123	664.7	1.1894
170	1.820	684.9	1.2641	1.638	682.5	1.2501	1.487	680.0	1.2369	1.359	677.5	1.2245	1.153	672.2	1.2014
180	1.859	691.3	1.2742	1.675	689.1	1.2604	1.521	686.7	1.2475	1.391	684.4	1.2354	1.183	679.5	1.2129
190	1.897	697.7	1.2840	1.710	695.5	1.2704	1.554	693.3	1.2577	1.422	691.1	1.2458	1.211	686.5	1.2239
200	1.935	703.9	1.2935	1.745	701.9	1.2801	1.587	699.8	1.2677	1.453	697.7	1.2560	1.239	693.5	1.2344
210	1.972	710.1	1.3029	1.780	708.2	1.2896	1.619	706.2	1.2773	1.484	704.3	1.2658	1.267	700.3	1.2447
220	2.009	716.3	1.3120	1.814	714.4	1.2989	1.651	712.6	1.2867	1.514	710.7	1.2754	1.294	706.9	1.2546
230	2.046	722.4	1.3209	1.848	720.6	1.3079	1.683	718.9	1.2959	1.543	717.1	1.2847	1.320	713.5	1.2642
240	2.082	728.4	1.3296	1.881	726.8	1.3168	1.714	725.1	1.3049	1.572	723.4	1.2938	1.346	720.0	1.2736
250	2.118	734.5	1.3382	1.914	732.9	1.3255	1.745	731.3	1.3137	1.601	729.7	1.3027	1.372	726.5	1.2827
260	2.154	740.5	1.3467	1.947	739.0	1.3340	1.775	737.5	1.3224	1.630	736.0	1.3115	1.397	732.9	1.2917
270	2.189	746.5	1.3550	1.980	745.1	1.3424	1.805	743.6	1.3308	1.658	742.2	1.3200	1.422	739.2	1.3004
280	2.225	752.5	1.3631	2.012	751.1	1.3507	1.835	749.8	1.3392	1.686	748.4	1.3285	1.447	745.5	1.3090
290	2.260	758.5	1.3712	2.044	757.2	1.3588	1.865	755.9	1.3474	1.714	754.5	1.3367	1.472	751.8	1.3175
300	2.295	764.5	1.3791	2.076	763.2	1.3668	1.895	762.0	1.3554	1.741	760.7	1.3449	1.496	758.1	1.3257
320	2.364	776.5	1.3947	2.140	775.3	1.3825	1.954	774.1	1.3712	1.796	772.9	1.3608	1.544	770.5	1.3419
340	2.432	788.5	1.4099	2.203	787.4	1.3978	2.012	786.3	1.3866	1.850	785.2	1.3763	1.592	782.9	1.3576
360	2.500	800.5	1.4247	2.265	799.5	1.4127	2.069	798.4	1.4016	1.904	797.4	1.3914	1.639	795.3	1.3729
380	2.568	812.5	1.4392	2.327	811.6	1.4273	2.126	810.6	1.4163	1.957	809.6	1.4062	1.686	807.7	1.3878

Properties of Saturated Ethylene*

Temp., °F.	Abs. pressure, atm.	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./ (lb.)°R.	
		Liquid v _f	Vapor v _g	Liquid h _f	Vapor h _g	Liquid s _f	Vapor s _g
-272.47	0.0012	4064.0		-68.19	176.5	-0.2826	1.024
-260.0	0.0037	1405.0		-60.84	180.1	-0.2351	0.972
-240.0	0.0177	328.6		-49.16	185.7	-0.1887	0.880
-220.0	0.0606	103.1		-37.56	191.4	-0.1382	0.817
-200.0	0.169	39.74		-26.03	197.1	-0.0922	0.767
-180.0	0.402		17.92	-14.57	202.6	-0.0498	0.727
-160.0	0.837		9.047	-5.12	207.2	-0.0103	0.692
-154.66	1.0000	0.02818	7.6712	0.0	207.9	0.0000	0.6814
-140.00	1.5775	0.02877	5.005	8.6	210.5	0.0249	0.6563
-120.00	2.7376	0.02964	2.987	20.8	214.2	0.0648	0.6340
-100.00	4.4616	0.03122	1.879	32.8	217.2	0.0995	0.6121
-80.00	6.8697	0.03179	1.732	45.2	219.9	0.1374	0.5974
-60.00	10.099	0.03308	0.857	57.9	221.9	0.1666	0.5769
-40.00	14.0338	0.03468	0.593	70.8	222.8	0.1935	0.5556
-20.00	19.722	0.03662	0.419	84.7	222.7	0.2245	0.5383
0.00	26.397	0.03912	0.301	100.3	221.0	0.2577	0.5202
+20.00	34.55	0.04292	0.212	119.6	216.6	0.2968	0.4991
+40.00	44.54	0.05035	0.139	148.7	206.2	0.3533	0.4683
+49.82	50.50	0.070	0.070	171.8	171.8	0.3964	0.3964

* York and White, Trans. Am. Inst. Chem. Engrs., 40, 227 (1944).

THERMODYNAMIC DATA

Properties of Superheated Ethylene*

v, volume, cu. ft./lb.; h, enthalpy, B.t.u./lb.; s, entropy, B.t.u./lb. (°R.)

Abs. pressure, atm.	Temperature, °F.																			
	-140°	-120°	-100°	-80°	-60°	-40°	-20°	0°	20°	40°	60°	80°	100°	120°	140°	160°	180°	200°	220°	240°
v	8.061	8.617	9.168	9.714	10.256	10.794	11.329	11.862	12.497	13.224	13.953	14.51	15.56	16.61	17.66	18.71	20.28	21.85	23.42	24.99
h	212.2	218.3	224.3	230.6	236.9	243.3	249.9	256.6	263.4	270.4	277.7	292.6	308.3	324.8	341.5	358.1	388.4	418.2	450.4	482.9
s	0.695	0.714	0.732	0.748	0.764	0.780	0.795	0.810	0.825	0.839	0.853	0.881	0.908	0.934	0.961	0.986	1.025	1.062	1.098	1.133
v	4.190	4.479	4.765	5.047	5.324	5.599	5.873	6.144	6.414	6.683	7.217	7.749	8.280	8.808	9.34	10.12	10.91	11.70	12.48	13.26
h	215.9	222.3	228.9	235.4	242.0	248.7	255.6	262.5	269.5	276.9	291.9	307.6	324.2	341.5	358.9	388.1	417.0	446.2	475.7	505.2
s	0.659	0.677	0.693	0.710	0.726	0.741	0.756	0.771	0.785	0.799	0.828	0.855	0.881	0.909	0.934	0.961	1.001	1.038	1.075	1.112
v	2.130	2.287	2.439	2.588	2.733	2.878	3.019	3.159	3.297	3.571	3.843	4.113	4.380	4.647	5.046	5.443	5.838	6.234	6.630	7.026
h	218.2	225.5	232.4	239.3	246.3	253.4	260.6	267.8	275.3	290.6	306.6	323.3	340.6	358.9	387.4	416.6	445.7	474.9	504.2	533.5
s	0.623	0.641	0.658	0.675	0.691	0.706	0.722	0.737	0.751	0.780	0.807	0.833	0.860	0.886	0.925	0.963	1.000	1.035	1.070	1.105
v	1.456	1.566	1.673	1.776	1.883	1.977	2.099	2.168	2.356	2.541	2.724	2.905	3.084	3.252	3.420	3.588	3.756	3.924	4.092	4.260
h	221.6	229.1	236.6	243.9	251.2	258.6	266.0	273.7	289.3	305.4	322.3	339.8	358.1	386.8	415.0	443.2	471.4	499.6	527.8	556.0
s	0.607	0.625	0.642	0.659	0.675	0.691	0.706	0.721	0.750	0.777	0.804	0.831	0.858	0.886	0.925	0.963	1.000	1.035	1.070	1.105
v	0.860	0.936	1.006	1.074	1.140	1.202	1.264	1.382	1.499	1.613	1.724	1.834	1.999	2.161	2.323	2.483	2.643	2.803	2.963	3.123
h	222.4	230.8	238.8	246.6	254.4	262.4	270.4	286.5	303.1	320.3	338.0	356.5	385.0	413.5	442.0	470.5	499.0	527.5	556.0	584.5
s	0.578	0.599	0.616	0.632	0.649	0.665	0.680	0.711	0.739	0.766	0.794	0.820	0.850	0.880	0.910	0.940	0.970	1.000	1.030	1.060
v	0.614	0.669	0.718	0.765	0.811	0.897	0.978	1.057	1.134	1.209	1.322	1.433	1.542	1.650	1.758	1.866	1.974	2.082	2.190	2.298
h	230.9	239.9	248.7	257.5	266.2	275.3	283.0	300.1	317.7	335.8	354.6	383.8	413.0	442.2	471.4	500.6	529.8	559.0	588.2	617.4
s	0.573	0.593	0.612	0.630	0.646	0.662	0.677	0.707	0.734	0.763	0.789	0.828	0.867	0.906	0.945	0.984	1.023	1.062	1.101	1.140
v	0.459	0.503	0.545	0.583	0.652	0.717	0.778	0.838	0.897	0.983	1.068	1.151	1.233	1.315	1.397	1.479	1.561	1.643	1.725	1.807
h	232.5	242.5	252.1	261.3	270.9	280.8	290.2	309.5	329.2	349.1	369.0	388.8	408.6	428.4	448.2	468.0	487.8	507.6	527.4	547.2
s	0.560	0.582	0.591	0.618	0.651	0.682	0.710	0.740	0.770	0.800	0.830	0.860	0.890	0.920	0.950	0.980	1.010	1.040	1.070	1.100
v	0.282	0.318	0.350	0.405	0.454	0.498	0.543	0.585	0.645	0.704	0.760	0.817	0.875	0.933	0.991	1.049	1.107	1.165	1.223	1.281
h	226.9	235.8	245.0	254.8	264.6	274.4	284.2	294.0	303.8	323.6	343.4	363.2	383.0	402.8	422.6	442.4	462.2	482.0	501.8	521.6
s	0.528	0.552	0.574	0.613	0.646	0.677	0.707	0.737	0.767	0.797	0.827	0.857	0.887	0.917	0.947	0.977	1.007	1.037	1.067	1.097
v	0.192	0.228	0.279	0.323	0.359	0.428	0.476	0.521	0.565	0.609	0.653	0.697	0.741	0.785	0.829	0.873	0.917	0.961	1.005	1.049
h	221.1	237.3	262.3	283.4	303.7	323.8	344.1	375.1	407.2	440.8	474.5	508.2	541.9	575.6	609.3	643.0	676.7	710.4	744.1	777.8
s	0.503	0.534	0.580	0.617	0.650	0.681	0.709	0.751	0.791	0.828	0.865	0.902	0.939	0.976	1.013	1.050	1.087	1.124	1.161	1.198
v	0.138	0.201	0.243	0.275	0.309	0.334	0.375	0.412	0.448	0.484	0.520	0.556	0.592	0.628	0.664	0.700	0.736	0.772	0.808	0.844
h	218.1	250.3	275.6	297.5	318.7	339.7	371.5	404.2	438.2	472.4	506.7	541.0	575.3	609.6	643.9	678.2	712.5	746.8	781.1	815.4
s	0.484	0.550	0.590	0.626	0.659	0.688	0.731	0.771	0.810	0.848	0.886	0.924	0.962	1.000	1.038	1.076	1.114	1.152	1.190	1.228
v	0.069	0.144	0.186	0.219	0.248	0.273	0.307	0.340	0.371	0.402	0.433	0.464	0.495	0.526	0.557	0.588	0.619	0.650	0.681	0.712
h	171.0	238.9	267.4	291.2	313.6	335.4	367.8	401.4	435.8	470.3	504.8	539.3	573.8	608.3	642.8	677.3	711.8	746.3	780.8	815.3
s	0.390	0.517	0.568	0.607	0.640	0.671	0.716	0.756	0.795	0.833	0.871	0.909	0.947	0.985	1.023	1.061	1.099	1.137	1.175	1.213
v	0.051	0.082	0.121	0.150	0.174	0.195	0.224	0.250	0.275	0.301	0.326	0.351	0.376	0.401	0.426	0.451	0.476	0.501	0.526	0.551
h	146.4	205.0	248.3	277.6	302.8	326.4	360.5	395.3	430.8	466.1	501.4	536.7	571.9	607.2	642.5	677.8	713.1	748.4	783.7	819.0
s	0.340	0.477	0.522	0.569	0.608	0.641	0.688	0.731	0.770	0.808	0.846	0.884	0.922	0.960	0.998	1.036	1.074	1.112	1.150	1.188
v	0.044	0.057	0.082	0.109	0.131	0.149	0.174	0.197	0.219	0.239	0.259	0.279	0.299	0.319	0.339	0.359	0.379	0.399	0.419	0.439
h	143.5	182.5	228.3	263.0	291.3	317.1	353.3	389.6	426.0	462.4	498.8	535.2	571.6	608.0	644.4	680.8	717.2	753.6	789.9	826.3
s	0.327	0.401	0.480	0.535	0.579	0.614	0.665	0.709	0.751	0.793	0.835	0.877	0.919	0.961	1.003	1.045	1.087	1.129	1.171	1.213
v	0.039	0.046	0.056	0.067	0.081	0.094	0.111	0.128	0.143	0.159	0.174	0.189	0.204	0.219	0.234	0.249	0.264	0.279	0.294	0.309
h	139.1	167.7	199.8	235.0	267.2	296.5	336.5	376.0	415.8	455.6	495.4	535.2	575.0	614.8	654.6	694.4	734.2	774.0	813.8	853.6
s	0.308	0.361	0.417	0.474	0.524	0.564	0.619	0.668	0.712	0.756	0.800	0.844	0.888	0.932	0.976	1.020	1.064	1.108	1.152	1.196
v	0.037	0.042	0.047	0.054	0.062	0.070	0.085	0.097	0.108	0.119	0.129	0.139	0.149	0.159	0.169	0.179	0.189	0.199	0.209	0.219
h	138.3	163.8	191.1	221.5	252.3	281.7	323.4	364.7	405.9	447.2	488.5	529.8	571.1	612.4	653.7	695.0	736.3	777.6	818.9	860.2
s	0.295	0.342	0.392	0.440	0.487	0.529	0.585	0.636	0.682	0.724	0.766	0.808	0.850	0.892	0.934	0.976	1.018	1.060	1.102	1.144
v	0.035	0.039	0.043	0.049	0.054	0.059	0.069	0.078	0.088	0.099	0.109	0.119	0.129	0.139	0.149	0.159	0.169	0.179	0.189	0.199
h	138.2	161.8	187.5	215.1	243.9	272.3	314.1	355.8	397.1	438.5	479.8	521.1	562.4	603.7	645.0	686.3	727.6	768.9	810.2	851.5
s	0.286	0.331	0.376	0.419	0.462	0.503	0.561	0.612	0.659	0.702	0.745	0.788	0.831	0.874	0.917	0.960	1.003	1.046	1.089	1.132
v	0.034	0.038	0.041	0.045	0.049	0.054	0.061	0.069	0.077	0.083	0.089	0.095	0.101	0.107	0.113	0.119	0.125	0.131	0.137	0.143
h	137.9	161.4	186.3	212.1	238.7	265.9	307.7	349.4	391.2	432.0	472.8	513.6	554.4	595.2	636.0	676.8	717.6	758.4	799.2	840.0
s	0.277	0.322	0.364	0.404	0.446	0.484	0.541	0.593	0.640	0.684	0.728	0.772	0.816	0.860	0.904	0.948	0.992	1.036	1.080	1.124

* York and White, Trans. Am. Inst. Chem. Engrs., 40, 227 (1944).

Properties of Saturated Methane*

Temp., °F.	Abs. pressure, lb./sq. in. <i>p</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy B.t.u./lb. (°R.)		Temp., °F.	Abs. pressure, lb./sq. in. <i>p</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy B.t.u./lb. (°R.)	
		Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>			Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>
-280	4.90	0.03635	24.04	0	228.2	0	1.2699	-180	191.5	0.04575	773	87.8	257.0	0.3767	0.9816
-270	8.44	0.03698	14.61	8.2	232.3	0.0423	1.2236	-170	240.0	0.04745	610	98.0	257.2	0.4127	0.9622
-260	13.80	0.03766	9.31	16.6	236.4	0.0823	1.1830	-160	297.0	0.04944	483	108.7	256.5	0.4476	0.9411
-250	21.71	0.03839	6.13	25.0	240.3	1.201	1.1468	-150	364	0.05197	381	120.3	254.5	0.4839	0.9169
-240	32.4	0.03915	4.24	33.3	243.9	1.578	1.1164	-140	440	0.05224	3008	133.2	251.2	0.5214	0.8903
-230	46.4	0.03999	3.04	42.0	247.3	1.962	1.0900	-130	527	0.05999	2318	148.1	245.9	0.5656	0.8622
-220	64.5	0.04092	2.23	50.0	250.2	2.333	1.0660	-120	627	0.06961	1613	171.8	231.4	0.6329	0.8088
-210	87.6	0.04193	1.67	59.5	252.8	2.693	1.0434	-115.8	673	0.0983	0.983	203.4	203.4	0.7232	0.723
-200	115.7	0.04306	1.281	68.8	254.8	3.062	1.0224								
-190	150.0	0.04431	0.990	78.2	256.2	3.419	1.0019								

Properties of Superheated Methane*
 v , volume, cu. ft./lb.; h , enthalpy, B.t.u./lb.; s , entropy, B.t.u./lb. (°R.)
 Parenthetic figures after pressures are saturation temperatures

Temp., °F. <i>t</i>	10 lb./sq. in. abs. (-26.6°F.)			20 lb./sq. in. abs. (-21.8°F.)			30 lb./sq. in. abs. (-24.2°F.)			40 lb./sq. in. abs. (-23.4°F.)			60 lb./sq. in. abs. (-22.2°F.)			80 lb./sq. in. abs. (-21.3°F.)			100 lb./sq. in. abs. (-20.5°F.)		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
-260	12.98	237.0	1.2262	7.04	245.8	1.1830	4.60	244.2	1.1273	3.75	253.6	1.1365	2.421	251.1	1.0775	1.954	259.5	1.0832	1.518	256.9	1.0476
-240	14.39	247.3	1.2750	7.76	256.3	1.2313	5.09	254.9	1.1767	4.12	264.3	1.1826	2.678	262.1	1.1258	2.153	270.7	1.1269	1.684	268.4	1.0935
-220	15.78	257.5	1.3214	8.47	266.6	1.2752	5.57	265.4	1.2217	4.49	274.9	1.2237	2.934	273.0	1.1681	2.438	281.5	1.1662	1.847	279.6	1.1339
-200	17.15	267.7	1.3644	9.17	276.9	1.3148	6.06	275.9	1.2618	4.85	285.2	1.2607	3.184	283.5	1.2062	2.725	290.9	1.2055	2.155	289.4	1.1701
-180	18.52	277.9	1.4032	9.86	287.0	1.3507	6.53	286.1	1.2982	5.21	295.5	1.2944	3.429	293.9	1.2407	2.939	299.3	1.2396	2.301	297.6	1.2092
-160	19.87	287.9	1.4386	10.56	297.1	1.3836	6.99	296.3	1.3315	5.21	305.8	1.3252	3.670	304.4	1.2721	3.172	306.5	1.2782	2.483	304.6	1.2332
-140	21.28	297.9	1.4710	11.25	307.2	1.4138	7.45	306.4	1.3621	5.56	316.1	1.3541	3.91	314.9	1.3015	3.408	313.5	1.2935	2.619	311.2	1.2619
-120	22.62	308.0	1.5009	11.94	317.4	1.4422	7.91	316.7	1.3908	6.25	326.4	1.3816	4.14	325.2	1.3293	3.680	324.1	1.2918	2.844	322.9	1.2619
-100	23.97	318.1	1.5290	12.61	327.6	1.4693	8.37	327.0	1.4182	6.60	336.6	1.4079	4.37	335.6	1.3558	3.935	334.5	1.3186	3.082	333.4	1.2891
-80	25.33	328.1	1.5561	13.29	337.7	1.4953	8.83	337.0	1.4443	6.94	346.9	1.4328	4.60	345.9	1.3811	4.135	344.0	1.3440	3.279	344.0	1.3148
-60	26.69	338.2	1.5816	13.97	347.8	1.5200	9.28	347.3	1.4691	7.29	357.1	1.4567	4.83	356.3	1.4052	4.363	355.4	1.3683	3.461	354.5	1.3393
-40	28.02	348.3	1.6066	14.64	357.9	1.5437	9.74	357.5	1.4929	7.63	367.5	1.4798	5.06	366.7	1.4284	4.619	365.9	1.3917	3.646	365.2	1.3628
-20	29.36	358.5	1.6303	15.32	368.3	1.5667	10.19	367.9	1.5159	7.97	377.8	1.5020	5.30	377.0	1.4508	4.842	376.4	1.4142	3.831	375.7	1.3854
0	30.72	368.7	1.6531	16.00	378.6	1.5888	10.64	378.2	1.5380	8.31	388.6	1.5234	5.52	387.6	1.4723	5.039	387.0	1.4358	4.017	386.3	1.4072
20	32.06	378.9	1.6752	16.66	388.9	1.6101	11.10	388.6	1.5594	8.65	398.9	1.5440	5.75	398.3	1.4930	5.239	397.7	1.4566	4.203	397.0	1.4280
40	33.43	389.1	1.6964	17.34	399.5	1.6306	11.54	399.2	1.5800	8.99	409.6	1.5642	5.98	409.1	1.5133	5.408	408.4	1.4770	4.389	407.9	1.4485
60	34.73	399.9	1.7169	18.02	410.2	1.6507	12.00	410.9	1.6002	9.33	420.5	1.5843	6.21	420.1	1.5334	5.619	419.4	1.4971	4.571	418.9	1.4687
80	36.10	410.5	1.7370	18.70	421.1	1.6707	12.44	420.8	1.6202	9.66	431.4	1.6036	6.44	431.0	1.5529	5.842	430.3	1.5166	4.756	429.8	1.4882
100	37.44	421.4	1.7570	19.37	431.9	1.6900	12.90	431.6	1.6396	10.01	442.6	1.6221	6.66	442.0	1.5714	6.019	441.4	1.5351	4.941	440.9	1.5068
120	38.78	432.2	1.7763	20.05	442.9	1.7084	13.35	442.7	1.6580	10.34	453.7	1.6402	6.89	453.3	1.5895	6.236	452.7	1.5533	5.126	452.2	1.5250
140	40.13	443.2	1.7946	20.73	453.4	1.7265	13.79	453.9	1.6760	10.69	464.6	1.6582	7.12	464.6	1.6075	6.464	464.1	1.5714	5.310	463.7	1.5432
160	41.46	454.4	1.8127	21.40	464.5	1.7445	14.25	465.2	1.6940	11.03	476.7	1.6760	7.34	476.3	1.6254	6.650	475.8	1.5893	5.494	475.4	1.5611
180	42.80	465.7	1.8302	22.07	475.7	1.7622	14.70	476.9	1.7118	11.36	488.3	1.6936	7.56	487.9	1.6430	6.836	487.4	1.6069	5.678	487.1	1.5788
200	44.13	477.3	1.8484	22.74	486.8	1.7798	15.16	488.5	1.7294	11.70	500.3	1.7106	7.77	499.9	1.6600	7.021	499.5	1.6241	5.861	499.2	1.5960
220	45.47	488.9	1.8659	23.41	497.9	1.7968	15.61	499.0	1.7464	12.04	512.4	1.7276	8.02	512.0	1.6670	7.209	511.6	1.6411	6.043	511.3	1.6130
240	46.81	500.9	1.8829	24.08	509.0	1.8137	16.06	510.2	1.7633	12.37	524.6	1.7444	8.24	524.2	1.6938	7.396	523.9	1.6579	6.226	523.5	1.6299
260	48.15	512.9	1.8998	24.75	520.1	1.8305	16.50	521.8	1.7801	12.71	537.0	1.7609	8.46	536.7	1.7103	7.583	536.3	1.6744	6.409	536.0	1.6464
280	49.49	525.1	1.9166	25.42	531.4	1.8470	16.94	533.2	1.7966	13.04	549.6	1.7771	8.69	549.3	1.7266	7.770	549.0	1.6907	6.592	548.7	1.6627
300	50.83	537.6	1.9331	26.09	542.7	1.8633	17.39	544.8	1.8129	13.38	562.6	1.7934	8.91	562.2	1.7429	7.956	561.9	1.7070	6.775	561.6	1.6790
320	52.16	550.2	1.9493	26.75	554.0	1.8795	17.83	556.7	1.8291	13.72	575.5	1.8096	9.14	575.2	1.7589	8.146	574.9	1.7230	6.959	574.6	1.6951
340	53.50	563.0	1.9655	27.43	565.9	1.8955	18.28	568.7	1.8451	14.05	588.7	1.8252	9.36	588.5	1.7747	8.330	588.2	1.7389	7.141	587.8	1.7101
360	54.84	576.0	1.9815	28.10	577.8	1.9113	18.73	580.9	1.8609	14.38	602.2	1.8411	9.58	602.0	1.7906	8.514	601.7	1.7548	7.326	601.3	1.7269
380	56.18	589.2	1.9973	28.76	589.1	1.9113	18.73	580.9	1.8609	14.05	588.7	1.8252	9.36	588.5	1.7747	8.330	588.2	1.7389	7.141	587.8	1.7101
400	57.51	602.6	2.0132	28.76	602.5	1.9271	19.17	602.4	1.8768	14.38	602.2	1.8411	9.58	602.0	1.7906	8.514	601.7	1.7548	7.326	601.3	1.7269
420	58.85	616.1	2.0290	29.43	616.0	1.9429	19.62	615.8	1.8926	14.72	615.7	1.8569	9.81	615.5	1.8064	8.698	615.3	1.7706	7.510	614.9	1.7427
440	60.19	629.9	2.0446	30.10	629.8	1.9585	20.07	629.7	1.9082	15.05	629.6	1.8725	10.04	629.4	1.8221	8.882	629.1	1.7862	7.694	628.8	1.7583
460	61.53	643.9	2.0601	30.77	643.8	1.9741	20.51	643.7	1.9237	15.38	643.6	1.8880	10.26	643.4	1.8376	9.069	643.2	1.8018	7.878	642.9	1.7739
480	62.87	658.0	2.0755	31.44	657.9	1.9895	20.96	657.8	1.9391	15.72	657.7	1.9034	10.48	657.5	1.8530	9.254	657.3	1.8162	8.062	657.0	1.7893
500	64.20	672.4	2.0907	32.10	672.3	2.0047	21.40	672.2	1.9543	16.05	672.2	1.9186	10.70	672.0	1.8682	9.438	671.8	1.8325	8.246	671.5	1.8046

Temp., °F. <i>t</i>	150 lb./sq. in. abs. (-190.0°F.)			200 lb./sq. in. abs. (-178.2°F.)			300 lb./sq. in. abs. (-159.5°F.)			500 lb./sq. in. abs. (-132.9°F.)			800 lb./sq. in. abs.			1000 lb./sq. in. abs.			1500 lb./sq. in. abs.		
	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>	<i>v</i>	<i>h</i>	<i>s</i>
-180	1.052	262.4	1.0267	0.830	269.4	1.0240	0.553	271.1	1.0198	0.300	261.9	0.9197									
-160	1.172	274.5	1.0716	0.830	269.4	1.0240	0.553	271.1	1.0198	0.300	261.9	0.9197									
-140	1.283	286.3	1.1106	0.923	281.6	1.0655	0.624	284.7	1.0327	0.342	265.9	0.9313									
-120	1.391	297.7	1.1452	1.010	293.5	1.1016	0.624	284.7	1.0327	0.342	265.9	0.9313									
-100	1.495	308.9	1.1769	1.092	305.2	1.1346	0.687	297.5	1.0706	0.356	280.4	0.9715	0.1441	236.9	0.8141						
-80	1.597	319.9	1.2065	1.172	316.6	1.1654	0.747	309.9	1.1040	0.402	295.5	1.0139	0.1969	267.9	0.8992	0.1262	238.3	0.8118			
-60	1.695	330.7	1.2344	1.247	327.9	1.1941	0.802	321.9	1.1345	0.443	309.3	1.0500	0.2359	287.9							

THERMODYNAMIC DATA

Properties of Saturated Carbon Dioxide*†‡

Temp., °F.	Abs. pressure, lb./sq. in.	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./lb. (°R.)		Temp., °F.	Abs. pressure, lb./sq. in.	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./lb. (°R.)	
		Con- densed phase*	Vapor	Con- densed phase*	Vapor	Con- densed phase*	Vapor			Con- densed phase*	Vapor	Con- densed phase*	Vapor	Con- densed phase*	Vapor
-140	3.18	0.01008	24.320	-121.5	129.2	0.6065	1.3908	-20	214.9	0.01498	4.168	9.1	138.5	0.9430	1.2372
-120	8.90	0.01018	9.179	-116.0	132.0	0.6232	1.3636	-10	257.3	0.01532	3.472	13.9	138.7	0.9532	1.2303
-100	22.22	0.01032	3.804	-110.1	134.3	0.6403	1.3199	0	305.5	0.01570	2.904	18.8	138.9	0.9636	1.2247
-80	33.98	0.01040	2.525	-106.7	135.1	0.6499	1.3033	10	360.2	0.01614	2.437	24.0	138.7	0.9744	1.2188
-60	50.85	0.01048	1.700	-102.5	135.7	0.6607	1.2881	20	421.8	0.01663	2.049	29.4	138.3	0.9856	1.2127
-40	74.82	0.01059	1.162	-98.0	135.9	0.6724	1.2726	30	490.8	0.01719	1.722	35.4	137.8	0.9976	1.2067
-20	97.50	0.01059	0.815	-97.9	135.9	0.6725	1.2724	40	567.8	0.01787	1.444	41.7	136.7	1.0092	1.1994
0	118.2	0.01060	0.613	-97.9	135.9	0.6725	1.2724	50	653.6	0.01868	1.205	48.4	135.0	1.0218	1.1917
20	145.8	0.01066	0.509	-97.9	135.9	0.6725	1.2724	60	748.6	0.01970	0.994	55.5	132.1	1.0353	1.1826
40	177.8	0.01066	0.509	-97.9	135.9	0.6725	1.2724	70	853.4	0.02112	0.804	63.7	127.5	1.0500	1.1724
60	106.9	0.01066	0.509	-97.9	135.9	0.6725	1.2724	80	968.7	0.02370	0.606	73.9	118.7	1.0694	1.1555
80	106.9	0.01066	0.509	-97.9	135.9	0.6725	1.2724	87.8	1069.4	0.03454	0.345	97.0	97.0	1.1098	1.1098

* Above the solid line the condensed phase is solid; below the line it is liquid.

† "Refrigerating Data Book," 5th ed., American Society of Refrigerating Engineers, New York, 1942.

‡ $s_f = 1.0$ at 32°F.
 $h_f = 36.7$ at 32°F.

Properties of Superheated Carbon Dioxide*†

v, volume, cu. ft./lb.; h, enthalpy, B.t.u./lb.; s, entropy, B.t.u./lb. (°R.); p, absolute pressure, lb./sq. in.

p	-75°F.	-50°F.	0°F.	50°F.	100°F.	150°F.	200°F.	300°F.	400°F.	600°F.	800°F.	1000°F.	1200°F.	1400°F.	1600°F.	1800°F.
1.00	93.90	100.0	112.2	124.4	136.6	148.8	161.0	185.4	209.7	258.5	307.2	356.0	404.8	453.6	502.3	551.0
10.0	283.2	288.0	297.8	307.7	318.0	328.4	339.1	361.4	384.7	434.4	483.1	542.4	599.6	658.6	718.8	780.0
20.0	1.4772	1.4892	1.5112	1.5316	1.5506	1.5684	1.5852	1.6165	1.6451	1.6969	1.7423	1.7829	1.8197	1.8533	1.8838	1.9123
40.0	9.280	9.902	11.15	12.38	13.61	14.84	16.06	18.51	20.96	25.85	30.73	35.61	40.49	45.36	50.24	55.11
80.0	282.6	287.5	297.3	307.3	317.3	327.3	337.3	361.3	384.6	434.4	483.1	542.4	599.6	658.6	718.8	780.0
160.0	1.3733	1.3853	1.4073	1.4277	1.4467	1.4645	1.4813	1.5126	1.5412	1.5930	1.6384	1.6790	1.7158	1.7494	1.7799	1.8084
320.0	4.586	4.904	5.542	6.119	6.778	7.407	8.016	9.247	10.47	12.92	15.36	17.80	20.24	22.68	25.11	27.55
640.0	281.9	287.0	296.8	306.8	317.3	327.9	338.8	361.1	384.5	434.3	483.1	542.4	599.6	658.6	718.8	780.0
1280.0	1.3417	1.3538	1.3759	1.3964	1.4154	1.4332	1.4500	1.4813	1.5099	1.5617	1.6071	1.6477	1.6845	1.7181	1.7486	1.7771
2560.0	2.239	2.404	2.738	3.053	3.363	3.688	3.993	4.615	5.230	6.458	7.688	8.901	10.12	11.37	12.56	13.78
5120.0	280.6	285.9	295.8	305.9	316.5	327.4	338.4	360.9	384.3	434.2	483.0	542.4	599.6	658.6	718.8	780.0
10240.0	1.3088	1.3211	1.3435	1.3642	1.3834	1.4014	1.4184	1.4499	1.4787	1.5305	1.5759	1.6165	1.6533	1.6869	1.7174	1.7459
20480.0	1.154	1.335	1.498	1.657	1.828	1.982	2.298	2.608	3.226	3.839	4.448	5.060	5.670	6.281	6.887	7.492
40960.0	283.8	293.8	304.1	315.1	326.4	337.7	360.2	383.9	434.0	486.9	542.3	599.5	658.6	718.8	780.0	840.0
81920.0	1.2778	1.3044	1.3284	1.3490	1.3679	1.3855	1.4177	1.4468	1.4991	1.5446	1.5852	1.6220	1.6556	1.6861	1.7146	1.7416
163840.0	0.8665	0.9799	1.088	1.208	1.311	1.411	1.525	1.734	2.148	2.559	2.966	3.373	3.781	4.188	4.592	5.000
327680.0	1.2833	1.3086	1.3297	1.3488	1.3666	1.3993	1.4285	1.4808	1.5263	1.5669	1.6037	1.6373	1.6678	1.6963	1.7233	1.7492
655360.0	0.6305	0.7207	0.8033	0.8986	0.9760	1.139	1.297	1.610	1.918	2.224	2.530	2.836	3.141	3.445	3.749	4.053
1310720.0	289.7	300.4	312.1	324.4	336.3	359.1	383.1	433.6	486.6	542.2	599.5	658.6	718.8	780.0	840.0	900.0
2621440.0	1.2666	1.2928	1.3154	1.3350	1.3529	1.3857	1.4151	1.4675	1.5133	1.5539	1.5907	1.6243	1.6548	1.6833	1.7103	1.7363
5242880.0	0.4891	0.5652	0.6376	0.7125	0.7748	0.9075	1.035	1.287	1.534	1.779	2.024	2.269	2.513	2.757	3.001	3.245
10485760.0	287.7	298.6	310.6	323.4	335.6	358.5	382.7	433.4	486.5	542.2	599.5	658.6	718.8	780.0	840.0	900.0
20971520.0	1.2519	1.2805	1.3038	1.3239	1.3421	1.3753	1.4049	1.4574	1.5033	1.5439	1.5807	1.6143	1.6448	1.6733	1.7003	1.7273
41943040.0	0.3948	0.4614	0.5237	0.5886	0.6407	0.7532	0.8604	1.071	1.273	1.482	1.687	1.891	2.095	2.297	2.499	2.699
83886080.0	285.6	296.7	309.1	322.4	334.9	358.0	382.3	433.1	486.4	542.1	599.5	658.6	718.8	780.0	840.0	900.0
167772160.0	1.2395	1.2694	1.2940	1.3145	1.3330	1.3671	1.3963	1.4490	1.4948	1.5356	1.5724	1.6060	1.6365	1.6650	1.6925	1.7195
335544320.0	0.3563	0.4100	0.4636	0.5065	0.5495	0.6407	0.7532	0.8604	0.8556	1.021	1.186	1.349	1.513	1.676	1.838	1.999
671088640.0	294.0	306.9	320.9	333.9	357.1	381.6	432.8	486.2	542.0	599.4	658.6	718.8	780.0	840.0	900.0	960.0
1342177280.0	1.2562	1.2813	1.3029	1.3219	1.3390	1.3660	1.3957	1.4389	1.4848	1.5256	1.5624	1.5960	1.6265	1.6550	1.6825	1.7095
2684354560.0	0.2858	0.3341	0.3780	0.4171	0.4598	0.5693	0.7212	0.8502	0.9874	1.125	1.261	1.397	1.533	1.668	1.803	1.938
5368709120.0	291.2	304.6	319.4	332.8	356.3	381.0	432.5	486.0	541.9	599.4	658.5	718.7	779.9	840.0	900.0	960.0
10737418240.0	1.2436	1.2699	1.2925	1.3124	1.3475	1.3779	1.4307	1.4766	1.5174	1.5542	1.5878	1.6183	1.6468	1.6733	1.6998	1.7263
21474836480.0	0.2216	0.2652	0.3040	0.3358	0.4022	0.4633	0.5817	0.6950	0.8079	0.9201	1.032	1.142	1.255	1.367	1.479	1.591
42949672960.0	287.6	301.6	317.4	331.4	355.1	380.2	432.1	485.8	541.7	599.3	658.4	718.6	779.7	840.0	900.0	960.0
85899345920.0	1.2282	1.2559	1.2797	1.3006	1.3370	1.3681	1.4215	1.4675	1.5083	1.5451	1.5787	1.6092	1.6377	1.6652	1.6927	1.7202
171798691840.0	0.1772	0.2174	0.2513	0.2795	0.3374	0.3901	0.4912	0.5881	0.6832	0.7785	0.8733	0.9672	1.062	1.156	1.250	1.344
343597383680.0	283.9	298.7	315.4	330.0	354.0	379.4	431.7	485.5	541.5	599.2	658.3	718.7	779.9	840.0	900.0	960.0
687194767360.0	1.2148	1.2438	1.2687	1.2905	1.3281	1.3599	1.4138	1.4599	1.5007	1.5375	1.5711	1.6010	1.6301	1.6592	1.6883	1.7174
1374389534720.0	0.1452	0.1823	0.2123	0.2383	0.2898	0.3363	0.4250	0.5093	0.5921	0.6747	0.7571	0.8385	0.9202	1.0020	1.0840	1.1660
2748779069440.0	280.3	295.7	313.4	328.6	352.8	378.6	431.1	485.3	541.4	599.0	658.2	718.6	779.8	840.0	900.0	960.0
5497558138880.0	1.2020	1.2323	1.2583	1.2809	1.3198	1.3525	1.4071	1.4534	1.4942	1.5310	1.5646	1.5951	1.6236	1.6511	1.6786	1.7061
10995116277760.0	0.1196	0.1483	0.1712	0.2016	0.2489	0.3173	0.3812	0.4436	0.5060	0.5680	0.6292	0.6906	0.7520	0.8134	0.8748	0.9362
21990232555520.0	288.2	308.4	325.1	350.0	376.5	430.1	484.7	541.0	598.8	658.0	718.4	779.7	840.0	900.0	960.0	1020.0
43980465111040.0	1.2111	1.2391	1.2631	1.3041	1.3380	1.3935	1.4404	1.4812	1.5180	1.5516	1.5821	1.6106	1.6381	1.6656	1.6931	1.7206
87960930222080.0	0.1101	0.1310	0.1663	0.2106	0.2689	0.3471	0.4374	0.5349	0.6408	0.7555	0.8798	1.0037	1.1276	1.2515	1.3754	1.4993
175921860444160.0	303.4	321.6	347.1	374.1	401.1	437.5	484.0	530.8	577.6	624.4	671.2	718.0	764.8	811.6	858.4	905.2
351843720888320.0	1.2218	1.2472	1.2903	1.3258	1.3828	1.4302	1.4712	1.5080	1.5408	1.5696	1.5944	1.6152	1.6327	1.6475	1.6599	1.6699
703687441776640.0	0.1042	0.1356	0.1621	0.2096	0.2531	0.2953	0.3374	0.3789	0.4199	0.4599	0.4987	0.5365	0.5733	0.6091	0.6439	0.6787
1407374883553280.0	318.4	344.2	372.5	428.1	483.5	540.2	597.2	654.2	711.2	768.2	825.2	882.2	939.2	996.2	1053.2	1

Properties of Saturated Steam: Pressure Table*

Abs. pressure lb./sq. in. <i>p</i>	Temp., °F. <i>t</i>	Volume, cu. ft./lb.		Enthalpy, B.t.u./lb.		Entropy, B.t.u./lb. (°R.)		Internal energy, B.t.u./lb.	
		Liquid <i>v_f</i>	Vapor <i>v_g</i>	Liquid <i>h_f</i>	Vapor <i>h_g</i>	Liquid <i>s_f</i>	Vapor <i>s_g</i>	Liquid <i>u_f</i>	Vapor <i>u_g</i>
1.0	101.74	0.01614	333.6	69.70	1106.0	0.1326	1.9782	69.70	1044.3
2.0	126.08	0.01623	173.73	93.99	1116.3	0.1749	1.9200	93.98	1051.9
3.0	141.48	0.01630	118.71	109.37	1122.6	0.2008	1.8863	109.36	1056.7
4.0	152.97	0.01636	90.63	120.86	1127.3	0.2198	1.8625	120.85	1060.2
5.0	162.24	0.01640	73.52	130.13	1131.1	0.2347	1.8441	130.12	1063.1
6.0	170.06	0.01645	61.98	137.96	1134.2	0.2472	1.8292	137.94	1065.4
7.0	176.85	0.01649	53.64	144.76	1136.9	0.2581	1.8167	144.74	1067.4
8.0	182.86	0.01653	47.34	150.79	1139.3	0.2674	1.8057	150.77	1069.2
9.0	188.28	0.01656	42.40	156.22	1141.4	0.2759	1.7962	156.19	1070.8
10	193.21	0.01659	38.42	161.17	1143.3	0.2835	1.7876	161.14	1072.2
14.696	212.00	0.01672	26.80	180.07	1150.4	0.3120	1.7566	180.02	1077.5
15	213.03	0.01672	26.29	181.11	1150.8	0.3135	1.7549	181.06	1077.8
20	227.96	0.01683	20.089	196.16	1156.3	0.3356	1.7319	196.10	1081.9
25	240.07	0.01692	16.303	208.42	1160.6	0.3533	1.7139	208.34	1085.1
30	250.33	0.01701	13.746	218.82	1164.1	0.3680	1.6993	218.73	1087.8
35	259.28	0.01708	11.898	227.91	1167.1	0.3807	1.6870	227.80	1090.1
40	267.25	0.01715	10.498	236.03	1169.7	0.3919	1.6763	235.90	1092.0
45	274.44	0.01721	9.401	243.36	1172.0	0.4019	1.6669	243.22	1093.7
50	281.01	0.01727	8.515	250.09	1174.1	0.4110	1.6585	249.93	1095.3
55	287.07	0.01732	7.787	256.30	1175.9	0.4193	1.6509	256.12	1096.7
60	292.71	0.01738	7.175	262.09	1177.6	0.4270	1.6438	261.90	1097.9
65	297.97	0.01743	6.655	267.50	1179.1	0.4342	1.6374	267.29	1099.1
70	302.92	0.01748	6.206	272.61	1180.6	0.4409	1.6315	272.38	1100.2
75	307.60	0.01753	5.816	277.43	1181.9	0.4472	1.6259	277.19	1101.2
80	312.03	0.01757	5.472	282.02	1183.1	0.4531	1.6207	281.76	1102.1
85	316.25	0.01761	5.168	286.39	1184.2	0.4587	1.6158	286.11	1102.9
90	320.27	0.01766	4.896	290.56	1185.3	0.4641	1.6112	290.27	1103.7
95	324.12	0.01770	4.652	294.56	1186.2	0.4692	1.6068	294.25	1104.5
100	327.81	0.01774	4.432	298.40	1187.2	0.4740	1.6026	298.08	1105.2
110	334.77	0.01782	4.049	305.66	1188.9	0.4832	1.5948	305.30	1106.5
120	341.25	0.01789	3.728	312.44	1190.4	0.4916	1.5878	312.05	1107.6
130	347.32	0.01796	3.455	318.81	1191.7	0.4995	1.5812	318.38	1108.6
140	353.02	0.01802	3.220	324.82	1193.0	0.5069	1.5751	324.35	1109.6
150	358.42	0.01809	3.015	330.51	1194.1	0.5138	1.5694	330.01	1110.5
160	363.53	0.01815	2.834	335.93	1195.1	0.5204	1.5640	335.39	1111.2
170	368.41	0.01822	2.675	341.09	1196.0	0.5266	1.5590	340.52	1111.9
180	373.06	0.01827	2.532	346.03	1196.9	0.5325	1.5542	345.42	1112.5
190	377.51	0.01833	2.404	350.79	1197.6	0.5381	1.5497	350.15	1113.1
200	381.79	0.01839	2.288	355.36	1198.4	0.5435	1.5453	354.68	1113.7
250	400.95	0.01865	1.8438	376.00	1201.1	0.5675	1.5263	375.14	1115.8
300	417.33	0.01890	1.5433	393.84	1202.8	0.5879	1.5104	392.79	1117.1
350	431.72	0.01913	1.3260	409.69	1203.9	0.6056	1.4966	408.45	1118.0
400	444.59	0.0193	1.1613	424.0	1204.5	0.6214	1.4844	422.6	1118.5
450	456.28	0.0195	1.0320	437.2	1204.6	0.6356	1.4734	435.5	1118.7
500	467.01	0.0197	0.9278	499.4	1204.4	0.6487	1.4634	447.6	1118.6
550	476.94	0.0199	0.8424	460.8	1203.9	0.6608	1.4542	458.8	1118.2
600	486.21	0.0201	0.7698	471.6	1203.2	0.6720	1.4454	469.4	1117.7
650	494.90	0.0203	0.7083	481.8	1202.3	0.6826	1.4374	479.4	1117.1
700	503.10	0.0205	0.6554	491.5	1201.2	0.6925	1.4296	488.8	1116.3
750	510.86	0.0207	0.6092	500.8	1200.0	0.7019	1.4223	498.0	1115.4
800	518.23	0.0209	0.5687	509.7	1198.6	0.7108	1.4153	506.6	1114.4
850	525.26	0.0210	0.5327	518.3	1197.1	0.7194	1.4085	515.0	1113.3
900	531.98	0.0212	0.5006	526.6	1195.4	0.7275	1.4020	523.1	1112.1
950	538.43	0.0214	0.4717	534.6	1193.7	0.7355	1.3957	530.9	1110.8
1000	544.61	0.0216	0.4456	542.4	1191.8	0.7430	1.3897	538.4	1109.4
1100	556.31	0.0220	0.4001	557.4	1187.8	0.7575	1.3780	552.9	1106.4
1200	567.22	0.0223	0.3619	571.7	1183.4	0.7711	1.3667	566.7	1103.0
1300	577.46	0.0227	0.3293	585.4	1178.6	0.7840	1.3559	580.0	1099.4
1400	587.10	0.0231	0.3012	598.7	1173.4	0.7963	1.3454	592.7	1095.4
1500	596.23	0.0235	0.2765	611.6	1167.9	0.8082	1.3351	605.1	1091.2
2000	635.82	0.0257	0.1878	671.7	1135.1	0.8619	1.2849	662.2	1065.6
2500	668.13	0.0287	0.1307	730.6	1091.1	0.9126	1.2322	717.3	1030.6
3000	695.36	0.0346	0.0858	802.5	1020.3	0.9731	1.1615	783.4	972.7
3206.2	705.40	0.0503	0.0503	902.7	902.7	1.0580	1.0580	872.9	872.9

* Abridged from Keenan and Keyes, "Thermodynamic Properties of Steam," Wiley, New York, 1936. Copyright, 1937, by Joseph H. Keenan and Frederick G. Keyes.

THERMODYNAMIC DATA

Properties of Superheated Steam*

r, volume, cu. ft./lb.; *h*, enthalpy, B.t.u./lb.; *s*, entropy, B.t.u./(lb.)(°R.).

Abs. pressure, lb./sq. in. (sat. temp.)		Temp., °F.												
		200	300	400	500	600	700	800	900	1000	1100	1200	1400	1600
1	<i>h</i>	392.6	452.3	512.0	571.6	631.2	690.8	750.4	809.9	869.5	929.1	988.7	1107.8	1227.0
(101.74)	<i>s</i>	2.0512	2.1153	2.1720	2.2233	2.2702	2.3137	2.3542	2.3923	2.4283	2.4625	2.4952	2.5566	2.6137
5	<i>h</i>	78.16	90.25	102.26	114.22	126.16	138.10	150.03	161.95	173.87	185.79	197.71	221.6	245.4
(162.24)	<i>s</i>	1.8718	1.9370	1.9942	2.0456	2.0927	2.1361	2.1767	2.2148	2.2509	2.2851	2.3178	2.3792	2.4363
10	<i>h</i>	38.85	45.00	51.04	57.05	63.03	69.01	74.98	80.95	86.92	92.88	98.84	110.77	122.69
(193.21)	<i>s</i>	1.7927	1.8595	1.9172	1.9689	2.0160	2.0596	2.1002	2.1383	2.1744	2.2086	2.2413	2.3028	2.3598
14.696	<i>h</i>	30.53	34.68	38.78	42.86	46.94	51.00	55.07	59.13	63.19	67.25	75.37	83.48
(212.00)	<i>s</i>	1.8160	1.8743	1.9261	1.9734	2.0170	2.0576	2.0958	2.1319	2.1662	2.1989	2.2603	2.3174
20	<i>h</i>	22.36	25.43	28.46	31.47	34.47	37.46	40.45	43.44	46.42	49.41	55.37	61.34
(227.96)	<i>s</i>	1.7808	1.8396	1.8918	1.9392	1.9829	2.0235	2.0618	2.0978	2.1321	2.1648	2.2263	2.2834
40	<i>h</i>	11.040	12.628	14.168	15.688	17.198	18.702	20.20	21.70	23.20	24.69	27.68	30.66
(267.25)	<i>s</i>	1.6994	1.7608	1.8140	1.8619	1.9058	1.9467	1.9850	2.0212	2.0555	2.0883	2.1498	2.2069
60	<i>h</i>	7.259	8.357	9.403	10.427	11.441	12.449	13.452	14.454	15.453	16.451	18.446	20.44
(292.71)	<i>s</i>	1.6492	1.7135	1.7678	1.8162	1.8605	1.9015	1.9400	1.9762	2.0106	2.0434	2.1049	2.1621
80	<i>h</i>	6.220	7.020	7.797	8.562	9.322	10.077	10.830	11.582	12.332	13.830	15.325	16.820
(312.03)	<i>s</i>	1.6791	1.7346	1.7836	1.8281	1.8694	1.9079	1.9442	1.9787	2.0115	2.0721	2.1303	2.1885
100	<i>h</i>	4.937	5.589	6.218	6.835	7.446	8.052	8.656	9.259	9.860	11.060	12.258	13.456
(327.81)	<i>s</i>	1.6518	1.7085	1.7581	1.8029	1.8443	1.8829	1.9193	1.9538	1.9867	2.0484	2.1056	2.1628
120	<i>h</i>	4.081	4.636	5.165	5.683	6.195	6.702	7.207	7.710	8.212	9.214	10.213	11.212
(341.25)	<i>s</i>	1.6287	1.6869	1.7370	1.7822	1.8237	1.8625	1.8990	1.9335	1.9664	2.0281	2.0854	2.1426
140	<i>h</i>	3.468	3.954	4.413	4.861	5.301	5.758	6.172	6.604	7.035	7.895	8.752	9.609
(353.02)	<i>s</i>	1.6087	1.6683	1.7190	1.7645	1.8063	1.8451	1.8817	1.9163	1.9493	2.0110	2.0683	2.1255
160	<i>h</i>	3.008	3.443	3.849	4.244	4.631	5.015	5.396	5.775	6.152	6.906	7.656	8.406
(363.53)	<i>s</i>	1.5908	1.6519	1.7033	1.7491	1.7911	1.8301	1.8667	1.9014	1.9344	1.9962	2.0535	2.1107
180	<i>h</i>	2.649	3.044	3.411	3.764	4.110	4.452	4.792	5.129	5.466	6.136	6.804	7.472
(373.06)	<i>s</i>	1.5745	1.6373	1.6894	1.7355	1.7776	1.8167	1.8534	1.8882	1.9212	1.9831	2.0404	2.0977
200	<i>h</i>	2.361	2.726	3.060	3.380	3.693	4.002	4.309	4.613	4.917	5.521	6.123	6.725
(381.79)	<i>s</i>	1.5594	1.6240	1.6767	1.7232	1.7655	1.8048	1.8415	1.8763	1.9094	1.9713	2.0287	2.0860
220	<i>h</i>	2.125	2.465	2.772	3.066	3.352	3.634	3.913	4.191	4.467	5.017	5.565	6.113
(389.86)	<i>s</i>	1.5453	1.6117	1.6652	1.7120	1.7545	1.7939	1.8308	1.8656	1.8987	1.9607	2.0181	2.0755
240	<i>h</i>	1.9276	2.247	2.533	2.804	3.068	3.327	3.584	3.839	4.093	4.597	5.100	5.603
(397.37)	<i>s</i>	1.5319	1.6003	1.6546	1.7017	1.7444	1.7839	1.8209	1.8558	1.8889	1.9510	2.0084	2.0658
260	<i>h</i>	2.063	2.330	2.582	2.827	3.067	3.305	3.541	3.776	4.242	4.707	5.172
(404.42)	<i>s</i>	1.5897	1.6447	1.6922	1.7352	1.7748	1.8118	1.8467	1.8799	1.9420	1.9995	2.0569
280	<i>h</i>	1.9047	2.156	2.392	2.621	2.845	3.066	3.286	3.504	3.938	4.370	4.802
(411.05)	<i>s</i>	1.5796	1.6354	1.6834	1.7265	1.7662	1.8033	1.8383	1.8716	1.9337	1.9912	2.0486
300	<i>h</i>	1.7675	2.005	2.227	2.442	2.652	2.859	3.065	3.269	3.674	4.078	4.482
(417.33)	<i>s</i>	1.5701	1.6268	1.6751	1.7184	1.7582	1.7954	1.8305	1.8638	1.9260	1.9835	2.0409
350	<i>h</i>	1.4923	1.7036	1.8980	2.084	2.266	2.445	2.622	2.798	3.147	3.493	3.839
(431.72)	<i>s</i>	1.5481	1.6070	1.6563	1.7002	1.7403	1.7777	1.8130	1.8463	1.9086	1.9663	2.0237
400	<i>h</i>	1.2851	1.4770	1.6508	1.8161	1.9767	2.134	2.290	2.445	2.751	3.055	3.359
(444.59)	<i>s</i>	1.5281	1.5894	1.6398	1.6842	1.7247	1.7623	1.7977	1.8311	1.8936	1.9513	2.0087

* Abridged from Keenan and Keyes, "Thermodynamic Properties of Steam," Wiley, New York, 1936. Copyright, 1937, by Joseph H. Keenan and Frederick G. Keyes.

INDEX

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